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THE PRINCIPLES AND PRACTICE OF BOILER CONSTRUCTION

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THE

PRINCIPLES AND PRACTICE

OF

BOILER CONSTRUCTION

A MANUAL OF INSTRUCTION AND USEFUL
INFORMATION FOR PRACTICAL MEN

BY

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XI

PREFACE

About the year 1881 I was requested by a number of mechanics to give them some practical information about the strength and construction of steam boilers. After some hesitation I agreed to do so, in order to help a body of men whose opportunities for study were limited, and whose lack of knowledge was more their misfortune than their fault. I accordingly prepared a series of papers, which were first read before the Engineering Association of New South Wales and afterwards published in book form for distribution amongst those interested.

From the number of letters received at the time, the effort appears to have been fairly successful; but comparatively few copies were printed and the book was soon out of print. It was, however, suggested that it might be re-written in accordance with modern standard practice, and in 1894 I published the first edition of the present work, which was also quickly exhausted.

During more recent years the few copies which passed through the hands of the booksellers were eagerly bought at a heavy premium—as much as two pounds having been paid for a single copy—and I received many requests to re-issue the book. I at last consented, and entrusted the work of correcting its many errors and of bringing it into line with the latest regulations of the Board of Trade and Lloyd's Register to a fellow Member of the Institution of Mechanical Engineers. He has also added to the section on Modern Water Tube Boilers.

As stated in my earlier efforts, it must be distinctly understood that I claim no originality, and I hope that the book will be found useful and prove an incentive to the study of more scientific works.

X III

CONTENTS.

INTRODUCTION.

Comparatively few standard works on Boiler Construction are sufficiently practical to commend them to mechanical engineers and boilermakers. Many have been written by exceptionally clever men, but the information, although accurate and sound, is not conveyed simply enough to reach the class for which it is intended. To profitably study most of these books a man must be comparatively well educated—that is, he must know something more than the rudiments of algebra and geometry; for it is almost always taken for granted that every student can solve simple equations, resolutions of forces, etc. Unless an engineer or boiler-maker can do that and more, the formula to him is almost a dead letter; and if he cannot get someone to show him how the rules are applied to practical work, even the best books are of comparatively little value to him. On this point I speak feelingly, having on many occasions been completely stuck up in trying to understand the rules as expressed scientifically, and I have even now a lively recollection of past difficulties. Men of natural gifts and cultivated talent do not always make the best teachers, because, as a rule, they write in accordance with their own keenness of perception, which is far above the average; consequently they fire too high, miss their mark, and their labour is often lost. I have therefore taken special care to make everything as plain and simple as possible, and have used ordinary figures in conjunction with the usual method set down in other books.

Engineers, as a rule, know a great deal more about engines than boilers; in the past, at all events, they considered the boiler a secondary affair. In the shop many skilled workmen are content, or rather have to be content, with handling tools expertly, and if a practical engineer or boiler-maker be asked anything about the material, the strength or the arrangement of the riveted joints, the strains

to which parts are subjected, the strength of shells, furnaces, stays, flat surfaces, etc., it will generally be found that he has only very vague and uncertain knowledge. Of late, however, there has been a distinct improvement; in engine and boiler shops, and also afloat, a strong desire to learn has developed, and most of the men are now anxious to know something of the principles which govern their daily work. Personal contact with good mechanics has shown me that many of them are naturally gifted with capacity and perception, but from the force of circumstances never had a chance of cultivating their powers. Others again will tell you that, although anxious to learn, they do not know how or where to get the information, that the books are too expensive, and the majority of them written in a style unsuited to the average intelligence—and so on. Of course there is a considerable amount of truth in this; but it should be impressed on all that there is no royal road to learning, and that any man taking up this or any other mechanical subject must make up his mind to study long and patiently. Energy, perseverance, and a quiet determination not to be beaten are the levers to be used, whose power, if properly handled and persistently applied, will be multiplied many times to his advantage.

During recent years exceptional advancement has been made in the various branches of engineering, but in none has the progression been so marked as in boiler construction. Working pressures have been more than doubled owing to iron being practically discarded for boiler work, the punching machine for first-class work is out of date, hand riveting is doomed, so is local heating and flanging—in fact, boiler-shop practice has been completely revolutionised. For heavy standard work machinery does almost everything, while the quality of steel is greatly improved, and every plate, bolt, rivet, and stay is so minutely measured and carefully calculated that to design and construct a modern high-pressure boiler for either land or sea requires a large amount of knowledge, skill, and practical experience.

THE PRINCIPLES AND PRACTICE OF BOILER CONSTRUCTION

Tensile Strength.

If we take an inch square bar of what we call mild steel and subject it in the direction of its length to a steady strain that will break it, and this strain is = 62720lbs. (= 28 tons), then we say that the tensile breaking strength of this steel is 62720lbs. per sq. in., or 28 tons per sq. in. which is about the average strength of steel boiler plates. All shell plates, steam and water space stays, etc., are exposed, and have to bear a tensile stress when the boiler is at work.

Elongation.

In breaking our 1in. square steel bar it would stretch, or elongate, a certain amount before fracture took place, and this amount measured between two fixed points is termed its elongation. For example: If we put the bar in the testing machine and place two centre punch marks 10in. apart, then steadily test the bar to destruction, we will find it has stretched, say, 20 per cent. of its length—that is, the distance between the centre punch marks will now be 12in.; therefore we say this particular bar (one square inch in section) has an elongation of 20 per cent. All boiler steel must at least have this amount of elongation; if not, it is rejected as being too hard, which means it does not possess the required amount of ductility for the work it has to do, and would be liable to crack in flanging, or be unable to bear the racking strains of ordinary working conditions. Tested steel specimens generally give an elongation above 20, and often as high as 30 per cent. In testing there are always two

distinct kinds of elongation—one due to the general stretching of the bar, which is distributed more or less uniformly throughout the length; the other to what is called the local elongation, which takes place, and is very distinct, at the point of fracture.

Contraction of Area.

Our 1in. bar in elongating 20 per cent. of its length in 10in. must of necessity cause a considerable reduction in the original sectional area, and this reduction is also of two kinds—the first being more or less uniformly distributed over the entire length; the second being local, sudden, and of much greater extent at the point of fracture. The difference between the original square inch and the section at point of fracture is termed the "Contraction of Area."

The average contraction of area in steel boiler plates whose tensile strength is 28 tons is from 40 to 50 per cent. Thick plates generally show more contraction than thin ones, and there is usually a difference of from 5 to 10 per cent. less contraction across the grain than with it; but in practice this difference is not taken into account, and we may say that mild steel is just as strong one way as the other.

Shearing Strength.

When two plates are riveted together and a force steadily applied at right angles, or nearly at right angles, to the centre line of the rivet, and if that force be 23 tons, or 51520lbs., then assuming the rivet to be one square inch in section, its shearing strength would be expressed as 51520lbs. per sq. in. There is a very material difference between the tensile and shearing strength of a steel rivet of the same section, amounting on an average to five tons per square inch, due in some measure to the intense tensional stress developed in holding the plates together. This stress is intensified as the rivet cools, and also by the hammering it receives, these two causes having probably some effect in

disturbing the original position of the particles, and thereby weakening its structure. It should always be remembered that nearly all rivets have to bear two distinct strains—one of tension in binding the plates together; the other of shearing, applied at, or nearly at, right angles to centre line of rivet.

Limit of Elasticity.

If we put our 1in. bar in the testing machine we can apply a certain force which will stretch or elongate it a small amount, say $\frac{1}{64}$in. in 10in. If we remove that force and find that the bar has returned to its original length we are then *within* the elastic limit; but if we find the bar longer than the original 10in. then we have exceeded the elastic limit, because the force applied has resulted in putting a permanent set in it. Therefore we may define the limit of elasticity to be where the force applied is greater than the elastic power in the material to return to its original length; in short, where elasticity ends and permanent set begins. Speaking generally, the elastic limit of boiler steel is about one-half its tensile breaking strength. It may and does vary, but 14 tons per square inch is a fair average. An important point in connection with this elastic limit is the following:—The factor safety for the best work is 4½ to 1—that is, the various parts of the boiler are made 4½ times stronger than the working pressure—and assuming such pressure to be 100lbs. per sq. in., then its bursting pressure would be 450lbs.; but it should never be forgotten that it only takes 225lbs. to reach the elastic limit, and any excess of pressure over and above that results in permanent set and permanent injury to the material. Looking at it in this sense we have only a margin of 2¼ instead of 4½. Any boiler with a working pressure of 100lbs. would, when new, be pressed or tested to 200lbs., which is within 25lbs. of its limit of elasticity—quite near enough. In the opinion of many capable of judging, the margin between the cold water hydraulic test and the elastic limit should be considerably increased; this could be done by testing all new boilers

to not more than $1\frac{1}{2}$, or at most $1\frac{3}{4}$ times their working pressure.

Ductility and Toughness.

In the old days, when there was any particular patching to do, the foreman always kept a few pieces of "Low Moor" iron plates carefully stowed away and ready for any special work, and he did this because he knew from experience that it was the only material that possessed the necessary ductility and toughness suitable for the job. Now "Low Moor" is almost completely out of court, and we have in our so-called mild steel a material infinitely superior, stronger, sounder, tougher, and which when properly treated is in every case thoroughly reliable.

Strength of Iron Boiler Plates.

For boiler-making, iron is practically a thing of the past, still it is necessary to consider and explain its various qualities. Special test plates of various thicknesses have been made from time to time to ascertain their tension, elongation, and contraction of area, etc. The results of careful testing show a very considerable variation. Some of the specimens gave exceptionally high results—23 tons with the grain and 23 tons per sq. in. across it, with an elongation of 19 per cent. and 13 per cent. respectively, while the contraction of area was 29 per cent. with and 19 across the grain. Such iron is exceedingly rare, and even when specially made for testing purposes its quality is very uncertain. The average qualities of good boiler iron, which might be put into any specification, and which any builder could be reasonably expected to supply, are as follows:—For boilers:

47000lbs. sq. in. = 21 tons, in tension with the grain.
40000lbs. sq. in. = 18 tons, in tension across the grain.
Contraction of area—With the grain, 15 to 20 per cent.
Contraction of area—Across the grain, 8 to 12 per cent.
Elongation—With the grain, 15 to 20 per cent.
Elongation—Across the grain, 10 to 12 per cent.

This applies for plates from ⅜in. to ⅝in. thick. For thicker plates, from ¾in. up to 1¼in. the tests show a large variation and no two samples harmonise, the principal difference being that for thick plates the elongation and contraction of area are less, and for elongation may be averaged as follows:—

With the grain ..	..	11 per cent.
Across the grain ..	..	8 per cent.

Lowering the tensile strength generally, but not always. gives an increased contraction and elongation, and consequently a greater amount of ductility, whereas any material increase in the tensile strength, as a rule, decreases those qualities.

All test strips are usually made 10in. long between the centres, and where practicable the section is made equal to one square inch.

For round iron bars, as used for stays in steam and water spaces, the tensile strength is about the same as the plates, but the elongation is more, ranging from 15 to 25 per cent., while the contraction of area is from 30 to 40 per cent. Iron rivet bars also vary in quality, but good rivets have a tensile strength of about 25 tons, with an elongation of 25 and a contraction area of 45 per cent. Iron rivets in single shear have in the past been taken as being of equal strength to a corresponding section of the plate. Recent experiments, however, clearly prove they are not, and on an average the strength of an iron rivet exposed to shearing is reduced from two to three tons per square inch. The value of double shear is always taken at 1·75 times the single shear, but the results obtained from actual experiments are by no means uniform, the variation in some cases being considerable.

Mild Steel Boiler Plates.

The tensile strength of such plates varies and ranges from 23 to 32 tons per square inch. For shells the steel must not exceed 32 tons, and is seldom below 27 tons. For furnaces and

combustion chambers, and those parts exposed to the direct impact of the heat and flame, many builders put in steel of a lower tensile strength in order to increase the ductility of these particular parts, 25 and 26 tons being often used, but of late several of our leading builders—and especially for furnaces—have raised the strength to 30 tons per sq. in., and this appears to have been done for the special purpose of increasing the "collapsing co-efficient" (explained further on), so that they get a higher working pressure with a thinner plate. This, however, is a very doubtful thing to do, as practical working goes to prove that such plates are too hard to stand the strain and varying temperature of modern furnaces, and especially those working from 160 to 200lbs. per sq. in. The mean elongation of average boiler steel, say of 28 tons tensile, is about 25 per cent., and must, as before stated, never be less than 20. The cold bending test is that all plates should bend to a radius of 1½ times their thickness, and until their sides are parallel, the distance between the sides being not more than three times the plate's thickness; and for plates intended for furnaces and combustion chambers exposed to heat and flame, they must stand bending to the same extent after being heated cherry red, and when at that heat must be cooled suddenly in water whose temperature is 80° Fah. There is seldom any difficulty with the steel in such tests. It will stand that and much more; in fact, as a rule, you can, with thin plates, bend them cold under a steam hammer, and bring the sides together without any sign of fracture. Sometimes there may be a difference in the elongation and contraction of area of thick and thin plates, but nothing material. The strength across the grain may vary 5 per cent., but in practice this is disregarded.

Steel stays as used in steam and water spaces give results very similar to the plates. Steel rivets are slightly stronger in tension, and the contraction of area higher—from 50 to 60 per cent.—while the results of testing show that the rivet itself after being made is slightly stronger than the bar

from which it was manufactured. Its shearing strength is taken at 23 tons per square inch when in single shear, and when in double shear at 1·75 times the single shear.

Regarding the iron and steel used in boiler construction, the following will give a general idea of their many and varied qualities:—

The different iron ores as dug out of the earth are numerous, and vary in richness and purity. In their natural state they contain from 25 to over 70 per cent. of metallic iron, and after passing through the blast furnace are known as "cast iron."

Cast iron may be described as the chemical union of metallic iron with from 2 to 6 per cent. of carbon, 5 per cent. of silica, and a very small percentage of manganese, phosphorus, and sulphur.

Wrought iron is distinguished from cast iron by having much less carbon in it: instead of having from 2 to 6 per cent. it has considerably less than 1 per cent.; and, as will be shown, a very fractional addition or subtraction to or from the amount of carbon has a most material effect on its qualities—in fact, the distinction between iron and our so-called mild steel is practically the difference in the amount of carbon in the respective metals, and is less in iron than in steel.

Steel.—Wrought iron becomes steel almost imperceptibly, because the addition of a very small fraction of carbon entirely changes the nature of the material. Such addition strengthens and hardens it, but always at the expense of its ductility and resilience, which last means its spring or shock-resisting powers. When we speak of steel boiler plates we in a practical sense misuse the word, because to a mechanic the word "steel" suggests a material that will take a cutting edge, and if you ask any man with a shop training he will very soon tell you that anything that does not "*temper*" is not and cannot be steel.

In the opinion of many this word should be changed, and

"refined ingot iron," or some other equally appropriate name, substituted. It would be more practical and much more sensible if the difference between iron and steel was represented by a line drawn at the point where its percentage of carbon is sufficient to make it "*temper.*"

When steel has a proportion of carbon in it exceeding 0·5 per cent. it confers upon the material the property of becoming hard, and can be tempered. Consequently any workman would at once recognise it as steel, and make use of it accordingly; but any material in which the percentage of carbon prevents its becoming hard, and that will not temper, should be recognised as iron.

Below the temper limit (and all boiler steels are below it) the material contains from 0·2 down to 0·15 per cent. of carbon, and if heated and suddenly cooled this so-called steel will be found to be quite soft. This forms one of the most practical tests of its suitability for boiler construction.

Between 0.15 and 0·5 per cent. of carbon the quality ranges from the softest of mild steel to the hardest of tool steel, and when the amount exceeds 0·2 per cent. it is entirely useless for boiler steel—the material would be too hard and brittle, and therefore unreliable.

The amount of carbon which will permit steel being welded ranges from 0·15 to 0·6, but it will not weld if the carbon exceeds 1·0 per cent. The ordinary market steels for boilers are distinguished from iron by their freedom from impurities and by the completeness with which they have been refined, as well as by the proportion of carbon in them. Boiler steel may be said to be at its best when the amount of carbon in it is from 0·15 to 0·2 per cent., it being at the same time entirely free from other elements, except the required amount of manganese and silica. *A celebrated maker divides steel into four classes, and this division gives a very fair idea of its various qualities.

1st class.—Representing extra mild steel, in which the

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*The analysis of steel as given by various authorities shows great discrepancies in the percentage of carbon.

carbon is as low as 0·05 and not higher than 0.2. Tensile strength, 25 to 32 tons per square inch. Average elongation in 8 in., about 25 per cent. Such steel does not temper, and is the material used for boilers, ship plates, girders, wire, etc.

2nd class.—Mild steel. Carbon from 0·2 to 0·35 per cent. Tensile strength, from 32 to 38 tons per square inch. Average elongation in 8in., about 25 per cent. This steel will weld, and hardens a little. Used for railway axles, tyres, rails, guns, etc.

3rd class.—Hard steel. Carbon from 0·35 to 0·5 per cent. Tensile strength, from 38 to 46 tons per square inch. Average elongation in 8in., from 15 to 20 per cent. Difficult to weld, but may be tempered. Used for rails, special tyres, springs, guide bars of engines, spindles, hammers, etc.

4th class.—Extra hard steel. Carbon from 0·5 to 0·65 per cent. Tensile strength, from 46 to 51 tons per square inch. Average elongation in 8in., from 5 to 10 per cent. Does not weld, but may be strongly tempered. Used for delicate springs, files, chisels, saws, and the various cutting tools.

When considering the suitability and those physical properties on which its constructive value depends, it is well to remember that for boiler-making it is not exclusively a question of strength, but we should judge and select it for its toughness, malleability, and its power of endurance under the action of forces which may extend, compress, expand, or contract it, and know something of its behaviour under repeated loads, and of how it is affected by material changes of temperature. The introduction of Triple and Quadruple expansive engines has caused an enormous increase in the working pressure, and, as a natural consequence, the proper design and construction of suitable boilers has become a matter of supreme importance. Present practice has resulted in considerable saving of fuel, but with all our knowledge we have only managed to convert into useful work from 10 to 12 per cent. of the total heat in the coal, the

other 90 per cent. being lost. This teaches us the necessity for humility, and also shows how little has been accomplished.

It is the knowledge of this fact that has induced many clever men to turn their attention and talents to the construction of vessels which would be more economical and equally safe. But among all the numerous and latest contrivances for generating steam, there is nothing that stands prominently forward as being remarkable for either economy or efficiency, and we are at present practically dependent on the cylindrical high pressure Boiler, as at present made and used, and which, though far from perfect, may be considered, when properly designed and constructed, as being reliable and safe.

What we have to consider is the fundamental principles which govern its construction, and although many discrepancies may be found when comparing experimental data, still we have a number of well-known facts and laws laid down for our guidance, which when thoroughly understood and properly applied are invaluable in the construction of vessels containing high pressure steam.

In the treatment of Boiler strength, there are three things that claim our special attention:—

1st. The consideration of the strength of a cylinder as seen in the shell and furnace.

2nd. We have to investigate the strength of a sphere represented by cambered ends, superheaters, and Steam Domes.

3rd. The strength and stiffness of flat surfaces as seen in the combustion chambers, flat ends, steam and water spaces.

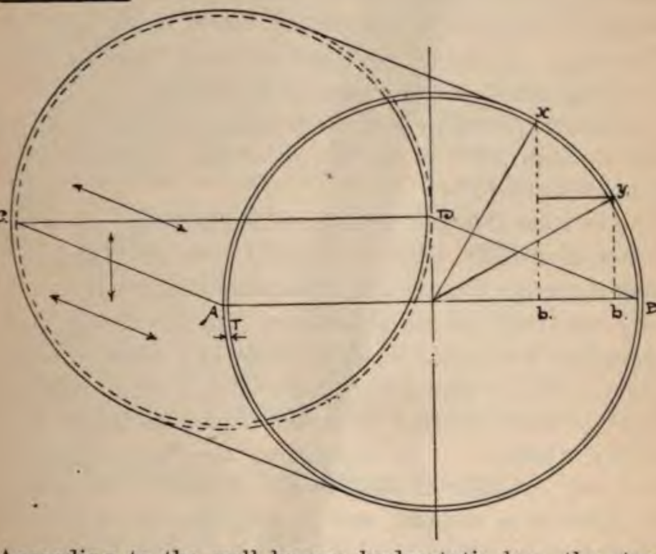
For the sake of making the explanation as simple as possible, it is assumed there are no riveted joints; they will be described and explained in due course.

Figure 1 is a sectional outline of a plain cylindrical Boiler, with flat ends and no stays; and here at the very start we meet a difficulty, viz., that to understand the action

of an internal force on a cylinder, it is necessary to know something of Trigonometry, as will be seen from the figure. However, as the scientific reason resolves itself into a very simple rule, which has been repeatedly proved to be correct, by actual experiment, we can accept and apply it with all confidence.

Referring again to the figure, we want to know how and in what direction the material is strained: the amount of force that would burst it, and how much steam pressure we should allow on the safety valve to make its working thoroughly safe.

Fig. 1 & 2.



According to the well known hydrostatic law, the steam pressure is exerted equally in all directions.

This pressure radiates in all directions from the centre, and the whole of the enclosing material is equally strained. The amount of pressure that would burst it—that is, divide it into two semi-circles, *in the direction of its length*—is represented by a force, acting and re-acting in opposite directions, which is resisted by the strength of the material.

At first sight, it would appear as if the pressure was acting over the whole of the semi-circumference and having a tendency to divide the shell in a plane drawn through the diameter.

It can be proved, however, that, the pressure being equal in all directions, the strain to which the half-circle is subjected is not to be taken as being equally effective in dividing the shell perpendicular to its diameter.

We are deeply indebted to Fairbairn for much valuable information on this subject. He was practically the original expounder of the true principles of boiler construction, and his method of demonstration, when calculating the bursting pressure, that the diameter and not the semi-circumference should be taken, is the one usually found in our text books.

Amongst the many explanations given, that by Wilson commends itself, and is somewhat as follows:—

In figure 1, consider first the pressure required to rend the shell through AB. It can be proved that the strain is only on the diameter CD.

As we leave this line, on either side the strain is less and is exerted diagonally, with less vertical force on A and B, and when it reaches these points it vanishes altogether.

By resolving the radial pressure into two forces, and by taking the component vertical forces at a large number of points, it can be shown that their sum is always equal to the boiler pressure exerted on a line equal in length to the boiler's diameter. In calculating the force required to burst any cylinder or shell in this direction, the length is not taken into account, and it simplifies the whole thing if we represent a portion of it by taking a ring, 1in. long, as it makes no difference, as will be shown presently, whether the cylinder be long or short in respect to the pressure required to separate the two sides.

Let ABCD represent the ring;

P, the pressure in lbs. per square inch;

xy, a very small portion of the circumference;

a, the angle xy makes with AB.

Then the pressure upon xy along the radius which passes through its centre will be P × xy.

If we decompose this force, the vertical component will be represented by P × xy × cosine of the angle; but xy × cosine of such angle is always equal to the projection bb of the arc xy on the diameter AB. The vertical component will then be equal to P × bb, and the sum of all the vertical components must of necessity be P × AB, the diameter of the boiler.

Now, all this elaborate and, to many practical mechanics, formidable explanation resolves itself into a very simple rule —viz., that in all cases the force of the steam tending to burst a cylindrical boiler is always equal to the internal diameter of the shell in inches, multiplied by the steam gauge pressure in lbs. per square inch.

As an example, assume that in a shell 72in. in diameter, we take a single ring 1in. long, and say the steam gauge shows 100lbs. per sq. in. Then 72 × 100 = 7200lbs. This is the strain the ring has to bear, and as the ring is 1in. long it is also the strain per square inch. That is, every square inch of internal surface has to bear a force of 7200lbs., and this force is equally divided between the two points in the line which forms the diameter of the boiler—that is, the points where it would burst, provided the steam strain was greater than the sectional strength of the ring. The truth of this has been repeatedly verified by actual experiment.

Length does not Affect Strength.

Take a piece of iron Boiler plate—*one inch square*—and assume that in tension a weight of 20 tons breaks it.

Take a second plate, of the same thickness, but 2in. broad; it will take 40 tons to break it, and if we take a third plate of the same thickness (1in.), but 3in. broad, it will take 60 tons to break it. It will be seen, therefore, that the strength of the 3 plates (being all the same thickness) depends entirely on their sectional area—one square inch

being good for 20, two square inches for 40, and three square inches for 60 tons respectively; and it must also be noted that the strain per square inch remains the same in all three plates, and is quite independent of the breadth.

If we now bend our 1in. square bar into a ring corresponding in thickness and diameter to the section of a Boiler-shell, the same reasoning holds good, as the following example will show.

Assume a shell to be 120in. internal diameter, and that the steam gauge pressure is 100lbs. per sq. in., then the strain on every linear inch of shell would be equal to the steam pressure in lbs. $\times$ diameter in inches—equivalent in this case to $100 \times 120 = 12000$lbs.; 2 inches would have to bear 24000lbs., 3 inches 36000lbs., 10 inches 120000lbs., and so on. But it will be very apparent that although the total strain increases directly as the length, the amount of material increases in the same ratio, so that the strain per square inch (as in the plate before it was bent into a ring) remains the same, for although in the present example a piece of shell plate 10in. long has to bear 120000 lbs., we have 10 square inches to do it, and as each inch takes $\frac{1}{10}$th of this, it is evident that in this sense the length of a Boiler does not affect its strength. There is one thing, however, in connection with the length not affecting the strength, that from a practical stand-point requires consideration, and it is this:—Although in all calculations the length is left out, because as we have seen the total pressure is always balanced by a proportional increase of material, the opinion is held by many that the ends do strengthen the shell very materially, especially in short Boilers; and where the length and diameter approach each other, it is considered that the shell strength is not far off that of a sphere of the same radius. This, however, is all assumption, but there is no doubt that in such a case the strength would be increased although perhaps not to the above extent.

Our knowledge on this point is limited, as experimental data on full-sized shells of different lengths, or, in fact, of

any length, are certainly rare, and in actual working the corrosion, grooving, flaws, wear and tear, have up to the present been considered sufficient reason for discarding any strength imparted by the ends. At the same time, if a series of experiments were carried out on full-sized shells, it is more than probable that our calculations would be considerably modified.

There is another thing relating to the internal force tending to burst a cylinder, viz., that in some cases increasing the thickness does not give a proportional increase of strength. This, of course, scarcely applies to boiler shells, but has an important influence on cylinders whose diameter and thickness are nearly equal, as seen in guns, hydraulic rams, etc. As an example, take a bar of iron one foot long, and assume a given weight stretches it $\frac{1}{8}$in.; then take another bar of the same sectional area, but eight feet long. The same weight will stretch it eight times as much, viz., 1in. And this proves that similar bars of the same quality and section when subjected to the same amount of tensile strain will, within certain limits, be stretched more or less uniformly in direct proportion to their length.

This being so, consider a hydraulic cylinder, and that its thickness is divided into a number of $\frac{1}{2}$in. rings. Say the inner ring is 12in. and the outer ring 48in. in circumference. Assume the material to be permanently injured if stretched $\frac{3}{32}$ in. per foot; then the inside ring would be damaged by a force that would expand the cylinder $\frac{1}{32}$ in. in diameter, while the outside ring, being four times as long, would only be stretched $\frac{1}{4}$th that amount.

In the shop are often seen fractured pipes and cylinders used for hydraulic purposes. Looking at the amount of broken section it appears strong enough for anything, and if the strength is increased in the same ratio as the thickness there would be an ample margin of safety over and above the working pressure. But from the above we know the strength does not increase as the thickness, or anything like it, so that adding to the thickness does no good, and it

would be better in many cases to double the area of the cylinder and work at half the pressure, which would give the same power with a much thinner cylinder. In boiler shells, however, the difference between the inside and outside circumference is so small compared with the diameter that the strain is practically considered to be uniformly distributed throughout the material.

From what has been said it is clear that the strain which all cylindrical boiler shells have to bear is always represented by the diameter in inches multiplied by the steam gauge pressure on the sq. in., and it is satisfactory to know that all the experiments of our eminent men agree to the following standard rule:—That the strength of cylinders to resist internal pressure is inversely as their diameter and thickness, from which it follows that the thickness of our shell plates must be in proportion to the diameter of the boiler.

The practical application of this rule is plainly seen when we take two boilers, one 60in. in diameter, the other 120in., working at the same pressure, say, 60lbs. per sq. in.

The strain on the 60in. shell would be $60 \times 60 = 3600$lbs., while on the 120in. shell we have $120 \times 60 = 7200$lbs., just double, and allowing the plates to be of the same quality and thickness, it is evident that if we have a margin of safety of 6 in the small boiler, we have only a margin of 3 in the large, and to make them proportional and equally safe we must do one of two things, viz., reduce the pressure in the large boiler to 30lbs., or double the thickness of the shell plates and work it at 60lbs.

Having explained the principle of an internal force acting on a boiler shell, let us inquire into the strength of the resisting iron or steel, as the case may be.

As an example: Take a boiler (still without seams) 72in. in diameter, the shell plates being ½in. thick, and the steam gauge pressure 100lbs. per sq. in. What have we got to resist this pressure?

Consider a single ring of this shell 1in. long; we have a

sectional area *on each side* equal to 1in. long $\times \frac{1}{2}$ in. thick $=$ $\frac{1}{2}$ a square inch *on each side.* Therefore, the total sectional area to resist the steam pressure is *one square inch.* The total steam pressure on each linear inch of shell is 72in. $\times$ 100lbs. $=$ 7200, and if we assume the boiler to be of iron, whose st·ength is 47000lbs. per square inch, we have a margin of safety of $\frac{47000}{7200} = 6\frac{1}{2}$—that is, the ring, or shell, is $6\frac{1}{2}$ times stronger than its working pressure.

If the boiler had been of steel whose strength was 28 tons $=$ 62720lbs. per sq. in., the margin of safety would be $\frac{62720}{7200}$ $= 8{\cdot}7$ times stronger than the working pressure.

It is very desirable and will give a much better grip of the subject if the above and other examples are worked out in more ways than one, because it often happens if you ask a mechanic to find the working or bursting pressure of any boiler, he can do it all right, but if he is asked to find any other factor, such as the strength of the iron, the diameter or thickness under certain given conditions, he generally makes a mess of it, because he has only studied the question from one point of view, and has, therefore, only a vague and a confused perception of the various relationships; and it will be a good investment and time well spent if he reverses the questions in various ways, so that with practice he may be able to find one factor just as easily as another.

To illustrate—Take the iron Boiler in the last example, and suppose the question is asked in the following manner:

In a boiler 72in. in diameter, with a working pressure of 100 lbs. per sq. in., and a factor of safety of $6\frac{1}{2}$, what must be its thickness? Here

$$\frac{\text{Diameter in in.} \times \text{pressure} \times \text{factor of safety}}{\text{Strength of the iron} \times 2} = \text{Thickness.}$$

And in this case

$$\frac{72 \times 100 \times 6\frac{1}{2}}{47000 \times 2} = {\cdot}5\text{in. thick}$$

If the boiler had been built of steel we get the same result:

$$\frac{72 \times 100 \times 8{\cdot}7}{62720 \times 2} = {\cdot}5\text{in thick}$$

Take another example: A modern steel boiler (without seams) is 15 feet in diameter, and the shell plates are 1in. thick. Would it be safe to work it at 155lbs. per sq. in.?

The strain on each linear inch of shell would be 180in. × 155 = 27900, and the material to resist this pressure is *one square inch on each side* = 2 *square inches altogether.* Therefore

$$\frac{62720 \times 2}{27900} = \frac{125440}{27900} = 4{\cdot}5$$

the factor of safety, so that it would be practically safe to work this boiler at 155lbs., because the Board of Trade have, under certain conditions, reduced the factor of safety for shells from 5 to $4\frac{1}{2}$. (This, however, would not be strong enough if there were riveted joints.)

Then to find the thickness when everything else is given we have

$$\frac{180 \times 155 \times 4{\cdot}5}{62720 \times 2} = 1\text{in.}$$

the required thickness.

The usual method of expressing the above in our text books is in the form of an equation, thus:

Let P = pressure in lbs. per square inch
,, D = diameter (internal) in inches
,, L = length in inches
,, T = thickness in inches
,, S = strength of material in lbs. per square inch.

$$\text{Then } P \times D \times L = 2 \times T \times L \times S$$

$$P \times D = 2 \times T \times S \qquad (1)$$

$$T = \frac{D \times P}{2 \times S} \qquad (2)$$

This expressed in words simply means (assuming the boiler to be on the point of bursting) that the pressure in lbs. × diameter in inches = sectional strength of the material (1); also, that the thickness, T, is always found by multiplying the diameter in in. by the pressure in lbs. and dividing by twice the strength of the iron.

This formula would apply to our 15 feet steel boiler in last example, as follows:—

The sectional strength of the steel was 4·5 times stronger than the steam pressure; therefore, to make

$$T = \frac{D \times P}{2 \times S}$$

we must raise the working pressure 4·5 times, and 155 × 4·5 =697lbs., the bursting pressure. Now, the diameter 180 × 697 = 125460lbs., and the material to resist this is 2 square inches = 62720 × 2, also = 125440lbs., so that

$$T = \frac{D \times P}{2 \times S}$$

proving that the force of the steam and the strength of the steel are in equilibrium, and that the slightest excess of pressure would blow the whole structure to pieces.

Oval Shells.

Although oval shells for high pressures are things of the past, it is desirable to point out that their behaviour under pressure is exceedingly complicated, as their resistance varies with every change of shape, and requires exceptional mathematical knowledge to investigate the strains.

There is one thing, however, in connection with oval shells which deserves our attention.

Boilers of this shape are generally made with top and bottom forming portions of a true circle, running into the flat sides, which are, or should be, stayed in accordance with the rules for flat surfaces.

Some mechanics are of opinion that any portion of circular plate is as strong as if it formed a part of a complete shell, but in many instances this is not so, and can only be the case when the ends of the circles, so to speak (where the cylindrical part joins the flat), are stayed with equal rigidity as they would be in a continuous circle, which would be represented by a tangential stress equal in force to the pressure multiplied by the radius.

Cases have occurred where, owing to the defective rigidity of the flat sides, the steam pressure actually put the top and bottom rows of stays in compression.

Hence in staying oval shells the principal consideration is the position and sectional area of cross stays, especially top and bottom rows, the object being to prevent any "spring" where the circular and flat surfaces meet.

Respecting the true strength of boiler material, it is necessary to point out that in all tests relating to iron or steel in tension *straight* bars, plates, or specially prepared strips are used in determining its resistance; consequently the material is only strained in one direction, whereas in a boiler shell it has to bear a lateral strain, being under stress longitudinally and transversely at the same time.

Having those forces practically pulling at right angles to each other, the question arose as to whether, under such conditions, the original tensile strength was injuriously affected, and it is satisfactory to know that practical experiments on wrought iron spheres conclusively proved that when strained simultaneously in all directions the iron is just as strong as when stretched in a straight line.

Another thing worth noting, and which specially applies to steel shells, is that if we set about trying to burst them, even with a hydraulic pump of unlimited power, we couldn't do it, because the material is so exceedingly ductile. When we remember that a steel strip 10in. long will extend, or elongate, $\frac{1}{4}$th of its length before it breaks, it must follow that if we subject any steel shell to excessive pressure the only effect will be to "oval" the rivet holes and let the water out; it would leak freely, but could not burst.

Transverse Strain.

We now come to the consideration of how a boiler shell is strained transversely, or circumferentially; that is the tendency to separate the shell into two rings, which would take place if a "ring seam" were to give way throughout its entire circumference.

The strain in this direction is very simply expressed, being always equal to the area of the end in square inches, multiplied by the pressure in lbs. per sq. in.

As an example, take a plain cylindrical iron boiler, 72in. internal diameter and ½in. thick, the working pressure being 130lbs. per sq. in. What amount of force would we have tending to blow the end out?

The area of a 72in. circle is 4071·5 square inches; therefore, 4071·5 × 130lbs. equals, roughly, 530000lbs., which is the total pressure acting against the end.

How much material have we got to resist this? An amount equal to the circumference of a 72in. cylinder multiplied by its thickness—in this case 72 × 3·1416 × ·5 = 113 square inches of material to hold the end.

Taking iron at 47000lbs. per sq. in., we have 47000 × 113 (roughly) = 5300000; so that dividing the *strength* by the *strain* we get

$$\frac{5300000}{530000} = 10$$

showing that the iron is 10 times stronger than the steam pressure.

But though we have a margin of strength of 10 to 1 in the direction of the "end" pressure, it can be shown and proved that the *same pressure* acting in the direction of the boiler's length only gives us a margin of safety of 5 to 1, instead of 10, because the pressure in lbs. multiplied by the diameter in inches, divided into the strength of the sectional thickness (1 square inch) = 5. Thus—72 × 130 = 9360, and

$$\frac{47000}{9360} = 5$$

showing that in all cylindrical shells, no matter what their diameter, thickness, or working pressure may be, the material in the direction of the boiler's length has to bear twice as much strain as the material in the direction of its circumference; that is, if it takes, say, 500lbs. per sq. in. to burst it in a fore and aft direction, it will take 1000lbs. to burst it athwart ships.

As another example showing the application of this principle to all cylindrical vessels, take the high pressure cylinder of a quadruple expansion engine.

Say it is 40in. in diameter and 1½in. thick, and that the total pressure is 200lbs. per sq. in. We want to prove that the strain in the direction of its length is twice as great as in the direction of its circumference.

The force tending to blow off the cylinder cover is equal to the area in the sq. in. multiplied by the pressure in lbs.—in this case, $1256 \cdot 6 \times 200 = 251320$lbs. ($= 112$ tons).

Then to resist this we have an amount of material equal to the circumference of a 40in. clinder multiplied by its thickness. Therefore, we have $40 \times 3 \cdot 1416 = 125 \cdot 7$ inches in length and 1½in. thick; so that $125 \cdot 7 \times 1 \cdot 5 = 188 \cdot 5$ square inches of sectional area to hold it.

Taking the tensile strength of cast iron at 7 tons per sq. in. $= 15680$lbs,. the total strength in our material must be $188 \cdot 5 \times 15680 = 2955680$lbs., and the total strain being 251320lbs., we have

$$\frac{2955680}{251320} = 12 \text{ (nearly), as the transverse}$$

margin of strength.

Looking at this cylinder the other way, longitudinally or lengthways, it has been shown in previous examples that its margin of strength is found by multiplying the diameter in inches by the pressure in lbs., and dividing by the sectional strength of the material. Applying this, and running out the figures, we have $40 \times 200 = 8000$lbs., the pressure per lineal inch.

To resist this we have twice the thickness, viz., 1½in. $\times 2 =$ 3 square inches; therefore $3 \times 15680 = 47040$lbs., the total strength: then dividing $\frac{\text{strength}}{\text{strain}}$ we get $\frac{47040}{8000} = 6$ as the margin of strength, which is just one-half, showing that the strains in all cylinders under pressure are as 2 to 1 in the direction of their length and circumference respectively.

In our text books the above is usually expressed as an

equation; but, before putting it in that form, it may be necessary to explain that in "formulæ" the symbol π always represents 3.1416 (the circumference of any circle whose diameter is 1), also that the $\frac{\text{Diameter}^2 \times \pi}{4}$ = the area of any circle, and is the same as the diameter$^2 \times \cdot 7854$ = the area of any circle.

Circumferential strain (using the same notation) is expressed thus: $P \times \frac{D^2 \times \pi}{4} = \pi \times T \times D \times S$.

To a mechanic who does not understand equations this method is, no doubt, confusing; but it simply means that when the pressure in lbs. × area of end in sq. in. = circumference in in. × thickness × strength per sq. in. the force of the steam and the sectional strength of the iron are exactly equal, and that the slightest excess of pressure would blow the end out.

Putting it practically, we might say that if the total pressure on end was 100 tons, and the total holding strength of the iron was also 100 tons, the boiler would be on the point of bursting.

To make this perfectly clear, let us apply this "formula" to a previous example, where the boiler was 72in. in diameter and ½in. thick, working at 130lbs., and where the margin of safety was found to be 10 and 5 to 1 respectively. As in this case we are dealing with the 10 to 1 margin, it is very evident if we increase the pressure 10 times, = 1300lbs., there will be no margin left; because 1300lbs. × 4071 (the area in sq. in.) = 5300000lbs. nearly, and sectional strength of the iron = 113 × 47000 = 5300000lbs. also: so that the strain and the strength are equal, or (what is the same thing) $P \times \frac{D^2 \times \pi}{4} = \pi \times T \times D \times S$, and this is what the formula means.

For the thickness in this (transverse) direction we have

$$T = \frac{P \times D}{4 \times S}$$

and as the formula for longitudinal strain is

$$T = \frac{P \times D}{2 \times S}$$

we again see, by comparing the two, that with the same pressure, diameter and thickness, a cylindrical shell is twice as strong fore and aft as it is athwartships, and that is why we use $4 \times S$ and $2 \times S$ as divisors.

As practical mechanics, however, we know that the above statements must be received with some caution; because when at work, and especially when getting up steam, in almost all boilers the ring seams, though theoretically twice as strong, are in many instances more severely strained than the longitudinal joints.

It has been previously shown that the elastic limit of iron is usually equal to one-half the breaking strength. Taking it at 10 tons per sq. in., and assuming that weight to produce a permanent set of $\frac{1}{8}$in., the very same amount of injury would be sustained by the iron if it was stretched to the same extent by increasing its temperature. Hence it is we have so much trouble when there is any considerable variation in the temperature of the various parts.

Iron expands when heated with a force exactly equal to the force with which it resists compression at the same temperature, and contracts with a force equal to the force with which extension to the same amount is resisted.

The co-efficient of expansion for iron plates not under pressure is ·0000064 of their linear dimensions; that is, an iron plate or bar will expand that extent for every additional degree of temperature. If under pressure, they alter in length according to the quality of the iron.

Rankine tells us that a strain of 186lbs. per sq. in. will give the same alteration in length as raising the temperature one degree Fah.

This is a wonderful statement, for allowing the ends of a plate to be rigidly fixed, an increase of one degree Fah. would subject it to compressive stress of 186lbs. per sq. in., while a decrease of one degree would ensure a tensile strain of

equal intensity, and those strains are quite independent of the sectional area of the iron.

It may be noted in passing that the words "stress" and "strain" are used indiscriminately. This is scarcely correct. Practically speaking, the word "stress" to a mechanic implies a force, or load, that the material may safely bear without injury; whereas the word "strain" is suggestive of a force, or load, which results in permanent set. This good old fashioned way is wrong, and the technical distinction, so far as our subject is concerned, might be expressed as follows:—Every boiler subjected to pressure changes its form, and this change of form is called the "strain," due to the pressure, or load. This strain may be within or beyond the elastic limit, but in all cases the molecular actions—that is, the alteration in the position of the particles of the steel or iron—developed by the steam pressure, which resist deformation, are called *"stresses."* These two words, however, are thoroughly "mixed" in our text books, and the above distinction is often disregarded.

Another thing relating to boiler ends, and to which attention might be directed, is that when internal furnaces are fitted the total pressure which the end plates have to bear is reduced in proportion to the size of the furnace. For example: Assume our 72in. boiler shell, ½in. thick, had a plain 36in. furnace, also ½in. thick, running through it, then the area of the boiler end exposed to the steam pressure is reduced ¼th, but the sectional strength is increased ½, because in any circle one-half the diameter of another the areas are as 1 to 4, but the circumferences are only as 1 to 2, consequently in this boiler shell, holding the end, we have 113 square inches + 56·5 square inches in the circumference of furnace, giving a total of 169·5 square inches, preventing end being blown out.

This shows that in all such cases the sectional strength of the furnace is to be added to the sectional strength of the shell plating, and also that where the furnace is one-half the diameter of the shell, and the same thickness, the holding

strength is increased 50 per cent., and the surface exposed to the steam pressure reduced 25 per cent.

Boiler-makers and engineers are well aware of the injurious effects of unequal expansion in shells and furnaces, and until lately few realised the great importance of providing means by which this great difficulty could be overcome, or, at all events, considerably lessened.

It is this subtile force which in almost all cases causes the leaks, ruptures, and distortions with which we have to contend—a force which does not show on the steam gauge or safety valve, and a force compared with which the steam pressure is often a very small fraction. An experiment showing this very plainly was tried many years ago, but where I forget. Two boilers were set in brickwork, but had separate furnaces under each. Sometimes the fire was only under one, but steam at 50lbs. per sq. in. filled both.

As water can only be heated by convection (circulation), and will scarcely receive heat downwards, the steam in this case was not condensed by the water in the boiler under which there was no fire, the bottom of which was practically cold. Both boilers were 30 feet long and 30 inches in diameter, without flues. The boiler under which the fire was had a temperature of 300° Fah. in the water and in the steam, and by careful measurement it was 1½in. longer than when cold.

The other having water in bottom half at 60° and steam in top half at 300° was by the force of expansion bent something like the segment of a circle, and commenced to leak badly along the bottom. This experiment justifies us in assuming that the strain on that boiler was just as severe as if the ends had been rigidly fixed and screwjacks applied underneath it, and worked up until the boiler was bent to the same extent.

Many accidents occur from this cause, and it may be (in the past) that the majority of explosions about which there was so much mystery could have been accounted for by the practically unknown forces which boilers have to bear from unequal expansion and contraction.

In designing, too much attention cannot be given to providing good circulation and the equalisation of temperature, especially when getting up steam, and although the means usually adopted to attain this are not altogether perfect, still they do lessen the evil materially.

On land, for internally-fired boilers, Galloway tubes in the furnaces and combustion chambers are the most effective, and should, where possible, be fitted; while in large marine boilers various plans are in use with varied success.

The "Hydrokineter" has given fair satisfaction, and, if used with care and judgment, when getting up steam the temperature at top and bottom of the boiler is practically the same. This can, and should always, be proved by the thermometer. Its action is very simple: Steam is taken from the donkey boiler and admitted by suitable cocks and cones to the main boiler near bottom of shell, where, parting with its heat, the surrounding water becomes heated, and is consequently lighter; it begins to move upwards, the colder water, being heavier, taking its place, and so the motion goes on until the whole of the water in the boiler is, as above stated, at the same temperature. This appliance, however, requires another boiler to generate the steam in the first instance, and this method is almost entirely confined to marine practice.

Another plan (open to the same objection) is to couple the donkey suction to boiler bottom, and pump the cold water from underneath the furnaces, discharging it at the water line. By continuous pumping for two or three hours the temperature at top and bottom is equalised.

Again, some makers fit very large vertical Galloway tubes in the combustion chambers, which also to some extent circulates the water from bottom to top, and this plan, being self-acting, commends itself to many; but in practice they have given great trouble, and in most cases have resulted in having to be cut out. This, however, is scarcely the fault of the tubes, but is often caused by bad design. Whenever such tubes are fitted the dimensions of the combustion chamber should exceed considerably the orthodox proportions. The

radius at the top and bottom flanges of the tubes should be as large and bold as possible, and special attention given to the top of the chambers to insure their not being stayed too rigidly, so that when heated the long tubes may have ample room to expand, for if due provision is not made for this an injurious compressive strain will be developed, resulting in cracking and wrinkling actions, which will destroy them in a short time. Several instances might be mentioned where after twelve years hard work such tubes are as good apparently as when they left the shop; but in a large majority of cases they have had to be cut out, and for the reasons mentioned above.

In all boilers where such appliances are not fitted one of the next best things to do is, when getting up steam, always fill the boiler well above the top gauge glass cock; then when water gets warm, and before any pressure shows, open, blow-off, and blow as much of the cold water out as possible; then when you get as much steam as will start donkey or injector, change water from bottom to top.

Collapsing Strain.

We now pass on to consider the strength of a furnace, also a cylinder, but having to bear quite a different strain to the shell or ends.

When under pressure the shell has to sustain a tensile or bursting strain; but the furnace is exposed to a collapsing pressure, which requires a distinct and different method of calculation.

In dealing with this important part of our subject we had for many years some well known "data" to guide us, which, although not perfect, were recognised as being reliable and safe. Their real strength under actual working conditions can scarcely be conclusively demonstrated; but the splendid and elaborate series of experiments lately carried out by the Board of Trade and Lloyds on full-sized furnaces of various

kinds has practically settled their respective strengths when under a cold water test, and the different collapsing "constants" are derived from the results of testing full-sized furnaces to destruction.

It is now over forty years since Fairbairn completed his valuable experiments on the collapsing strength of tubes. As before stated, it is to Fairbairn we are indebted for much information respecting boiler strength, although some of his conclusions in the light of later experiments have had to be considerably modified, still the work he did must always be recognised and appreciated as being invaluable.

One of the most remarkable results of his labours was that with the same diameter and thickness of furnace *the strength decreased as the length increased*, and from his experiments he constructed a "formula," which for many years was almost invariably received and adopted as *the* standard rule, and which was used by engineers and boiler-makers with all confidence.

What Fairbairn proved was this: That when furnaces are exposed to collapsing strain their strength was inversely as their length, inversely as their diameter, and as the square of their thickness, which simply means that if we take two furnaces the same diameter and thickness, only one is 5 and the other 10 feet long, the short one is twice as strong as the long one; or if we have two furnaces of the same length and thickness, only the one is 2 feet and the other 4 feet in diameter, then the small one is double the strength of the large one.

But all, or nearly all, Fairbairn's experiments were conducted with comparatively small, thin tubes, and did not embrace the conditions under which modern furnaces are constructed, which has resulted in considerable modification of his formula, and this is specially so in relation to the various forms as now used for the exceptionally high working pressures.

One of the most important objections to the application of Fairbairn's "formula" to modern boiler practice is that there is virtually only one rule for all kinds and classes of

construction, and no provision or sliding scale for punched or drilled holes, or for plates drilled or punched before or after bending, whether they are drilled in place or otherwise, or whether the material is disposed so that its maximum strength is economically utilised—in fact, no distinctive variation in the strength constants for good or bad work.

The absolute necessity of making rules, which shall apply to all classes of boiler work, has been fully recognised by all modern authorities, and the Board of Trade has no less than fifteen "constants," ranging (for iron) from 90,000 for the very best down to 60,000 for the very worst class of work.

For steel the number of constants is the same, ranging from 99,000 to 66,000, which means 10 per cent. more for steel than iron.

It must also be remembered that the above 30 strength constants refer and apply to *plain* furnaces alone, and that each patent corrugated and ribbed furnace has each a different and distinct constant of its own, which will be described and explained in due course.

That such distinctions should be made is very necessary, and they have already been the means of vastly improving design and workmanship, besides giving credit where it is due, and have proved a strong incentive to increased excellence in construction.

Fairbairn's rule is as follows:—

$$P = \frac{806300 \times T^{2.19}}{L \times D}$$

Where P = collapsing pressure per square inch
T = thickness in inches
L = length in feet
D = diameter in inches.

Instead of the 2·19th power the square of the thickness is usually allowed to be near enough in practice. Expressing the rule in words, we would say: "Constant" × square of the thickness ÷ length in feet × diameter in inches = the collapsing pressure.

Take as an example a furnace 8ft. long, 36in. in diameter, $\frac{3}{8}$in. thick, and lap jointed, we would have

$$\frac{806300 \times \cdot 375^2}{8 \times 36} = 392\text{lbs.}$$

the collapsing pressure, and allowing a factor of safety of 6 to 1, the working pressure would be $\frac{392}{6} = 65$lbs. per sq. in.

In very long furnaces they are strengthened by angle or T iron rings fitted in the usual way, the distance between the ring centres representing the furnace length, so that if the flue was 20 feet long a ring riveted round the centre was considered to be just as strong as if it were only 10 feet long.

The exact amount of strength imparted by such rings is very doubtful, and several surveyors have stated that under certain conditions, which often occur in practice, they actually weaken the furnace.

This, I apprehend, refers to allowing the space between the ring and furnace crown to become dirty, when, of course, the plate becomes overheated, and, as a natural consequence, buckles and cracks. But that would and does happen with any plates having no rings if they be neglected; but there is no doubt if rings are properly fitted, *kept clean,* and the furnace ends not too rigid, they add most materially to the strength, although not perhaps to the extent and proportion stated in the rule. However, it is satisfactory to know, as boilers were made in Fairbairn's time, they were practically safe for the pressures then carried.

In all furnaces under *steam* pressure there is often, in fact always, a great difference between top and bottom. The top being at a higher temperature will, of course, be longer, and if we assume the end fastenings to be perfectly rigid, the top would be *compressed* and the bottom stretched, and for every degree of difference of temeperature we would have a compressive strain on top and a tensile strain on bottom of 186 $\div 2 = 93$lbs. per square inch.

This is merely the theory of unequal expansion and contraction; but we know in practice that nothing of the kind takes place.

The furnace ends are not rigid, and in almost all cases, when ordinary care is exercised, they accommodate themselves to any moderate difference of temperature.

It is desirable to mention here that in Fairbairn's experiments the flues or furnaces were small in diameter, lap-jointed, and single riveted, and not more than the thickness of the plate off a true circle.

Another thing in connection with all furnaces is that we soon come to the limit so far as the square of the thickness is concerned.

We cannot, or, at all events, we do not care about increasing the thickness beyond $\frac{5}{8}$in. or $\frac{3}{4}$in. at most, as in the opinion of those capable of judging thicker plates could not transmit the heat quick enough to prevent the fire side of the plate being burnt. This is a question which is attracting considerable attention at the present time, and is still unsolved; but several valuable experiments by Dr. Kirk, Blechynden, Morrison, and others have been carried out lately (referred to further on), which show very clearly that there is a very considerable temperature difference between the fire and water side of furnace plates, and that such difference becomes greater as the plates are increased in thickness.

Referring again to Fairbairn, it is necessary to point out that his rule (for plain furnaces) still holds good so far as the length and thickness are concerned. It is the strength "constant" that has been materially altered, because the true strength has from time to time been conclusively proved by the testing to destruction of *full-sized furnaces.* The constants now used are all derived from the actual collapsing pressures, and it is most interesting to note how the value of the constants have increased as the strength of the material and workmanship improved.

All such experiments, however, have been carried out under the cold water hydraulic test, where the temperature was uniform, and not under the actual working conditions, when the variation in the temperature becomes a material and important factor in the calculation, and this must of neces-

sity remain so because of the difficulty and danger of obtaining collapsing results when under steam.

About the year 1880, when the working pressure rose from 60 to 100lbs., the thickness of our furnace plates had to be proportionately increased, and we are indebted to Samson Fox for the design and construction of a furnace which enabled us to work at a much higher pressure with a comparatively thin plate.

In 1878 Fox made his first corrugated flue, a sketch of which is shown in Figure 3, which also shows the testing apparatus used for determining the relative collapsing strength of plain and corrugated furnaces.

This was tried in March, 1878, at the Leeds Forge Company's works, in the presence of a large number of engineers and boiler builders.

Two iron furnaces were experimented upon under cold hydraulic pressure. One was corrugated, the other plain, and both made of the same quality of iron. They were practically of the same dimensions, being about 38in. in diameter, $\frac{3}{8}$in. thick, and 7ft. long.

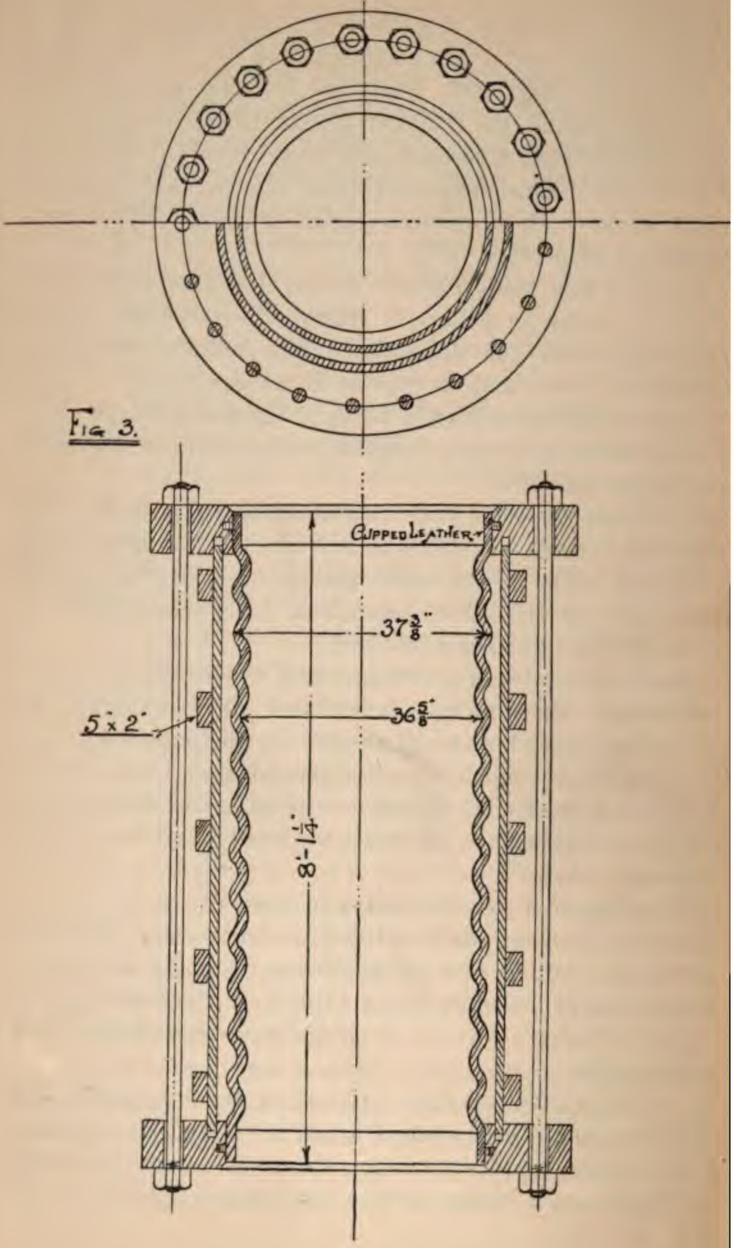
The test vessel was a wrought iron welded cylinder, 3in. larger inside than the tube to be tested. Five welded rings, 5in. x 4in., were bored and shrunk on, the ends being cast iron rings connected by longitudinal bolts, and bored to receive the turned ends of tubes, with a groove to receive a cupped leather washer, such as is commonly used for packing hydraulic rams.

Both furnaces were welded, and were fairly cylindrical. The plain furnace was tested first, and gave way at 200lbs. per sq. in., failing first at the horizontal diameter (least), the amount of deflection being $1\frac{1}{2}$in., and about 2ft. long.

By Fairbairn's formula it should have stood 350lbs. per square inch.

The plain tube was then replaced by the corrugated, and the pressure applied, when it failed at 450lbs., proving that under similar conditions, being the same length, diameter, and thickness, it was more than twice as strong as the plain one.

Fig. 3.



The result of such experiments gave corrugated furnaces a great impetus, for they not only demonstrated the superiority of the corrugations, but proved we could work at more than double the pressure with the same thickness of plate.

Another important matter in connection with them is, that their length does not affect their strength, as the length of a corrugated furnace is measured by the distance between the centres of the corrugations. Consequently, we have only to construct a formula which will limit the steam pressure in such a way as to prevent an excessive crushing strain being put upon the iron. This is effected by what we may term the "*limiting formula,*" the object being to prevent the material being subjected to a strain which would exceed its elastic limit, and so permanently injure it. This will be explained in due course when we come to consider the respective strengths of the different furnaces, each of which has a different constant.

In July, 1880, the first corrugated steel furnace was tested in a similar manner to the iron one described.

It was approximately of the same dimensions and stood a pressure of 600lbs. before it collapsed, but in this case more care was taken to make it more truly circular than in the first instance, being in no part more than its thickness off a true circle.

Various other experimental furnaces were tested, and in all it was noticed they gave way in the direction of their shortest diameter, and in the furnace that collapsed at 600 lbs. per sq. in., it was permanently bulged about 5in.

The cross section of the tube after failure could scarcely be called an oval, for it greatly resembled the cross section of an ordinary flat-sided boiler, with a semi-cylindrical top and bottom. The respective diameters were 36·06in. vertically and 31·12in. horizontally, the difference being 4·84in.

Although bulged inwards to that degree it did not even leak, and it was tested again in its altered condition, and it sustained a pressure of 350lbs per sq. in., when it collapsed almost as suddenly as in the first instance.

This is an important feature in a corrugated furnace, for it sometimes happens (but almost always through preventable causes), that they get out of shape in places to the extent of one, two, and even three inches, caused by accumulation of scale, dirt, or oil; but I have often tested such furnaces to double their working pressure, and there was practically no movement, and certainly no permanent set; and if the cause of such local flattening was removed—which means that the furnaces were cleaned and *kept clean*—such furnaces would work and have worked for years. Again, when partial flattening takes place at sea—and it may be thousands of miles from port—it gives the engineer confidence to complete his voyage, *after cleaning,* when the furnaces can be "set up" to their original shape.

Referring again to the first experiment, when the plain furnaces collapsed at 200lbs.—when by Fairbairn's rule it should have stood 350lbs.—much has been said and written, and any expression of opinion for or against experiments can only be taken for what it is worth.

First of all, it is evident that these and all such experiments, though of undoubted value, do not, and probably never can, embrace all the conditions relating to furnaces in actual work.

The cold water test is, beyond question, the best, and, in fact, the only means we have of determining their strength—but we do not take into consideration or make any allowance for the many racking and practically unknown strains which the furnace especially has to bear through sudden and material variations in the temperature; and there is little doubt that a furnace under steam is weaker than when under cold water pressure.

When furnaces are tested to, say, double their working pressure, if there is any movement, it will be perceptible in an alteration of the relative diameters at right angles to each other, which may or may not affect a large percentage of the length; and wherever the weakest spot is, there the diameter will be least. If we continue increasing the pressure (unless

any spot is exceptionally weak), its effect will produce a contrary distortion at right angles.

Up to a certain point, the strength of the tube may not be affected by its length; but, when distortion becomes general, it is then that the element of length comes in to affect this resistance; and it is then that the ends materially assist the furnace in sustaining the pressure—the material under those conditions being in a state of tension. And in this particular case it is the opinion of many engineers that if the ends of the plain iron furnace had been secured to ends in the test cylinder, the longitudinal tension upon the tube would have brought out the collapsing pressure—nearer Fairbairn's calculation, viz., 350lbs. per sq. in.

Again, in furnaces under pressure, and which have a tendency to flattening, that is, altering their respective diameters, the shorter the tube the sooner this flattening will be stopped by the assistance from the ends. And, as it has been already shown that the amount of stretching due to a given load is a factor of the length in plates of uniform section, it follows that the deflection and pressure tending to produce collapse depends, to some extent, on the length of the furnace.

Another thing is of considerable importance, and specially applies to long furnaces. If we assume the strength varies inversely as the length, Wilson shows that a furnace of the same dimensions as the experimental plain furnace—but 32 feet long—would be destroyed by a pressure of 50lbs. per sq. in., and it is well known that long unstrengthened tubes of similar dimensions, and of a weaker shape, have worked for years at this, and even a greater pressure. From this we may reasonably conclude that the ring seams must be credited with a value they have never received, and that in such furnaces the strength does not vary inversely as the length.

In all methods of calculating collapsing pressures, the ring seams get no credit. But there is no doubt, when such seams are continuous, that they do strengthen the furnace consider-

ably, although to what extent could only be determined by actual test. But in support of their adding to the strength, I have often, in fact, always noticed that when any furnace flattened, got out of shape, or came down in pockets, the extent of the injury was least at the seams and greatest in the plain spaces between them.

Modern Furnaces for High Pressures.

When Mr. S. Fox made his first corrugated flue in 1878, he had no special machinery to help him. The work was all done by hand, and the corrugations formed from a plain furnace—hammered into shape on a corrugated anvil block.

This method of manufacture, in the light of present appliances, was crude, and certainly not conducive to exact circularity; but, since that time, the machines invented for this special purpose have enabled furnaces to be constructed which are practically true circles.

A paper read by D. B. Morrison (the inventor of the Morrison Suspension Furnace), in 1892, before the North East Coast Institution of Engineers on "Marine Boiler Furnaces," gives a large amount of useful information. In fact, it is, without exception, by far the most valuable and certainly the most practical contribution ever published on the subject.

As the method of constructing modern furnaces may be of interest, the following extract from Morrison's paper will show how they are made:—

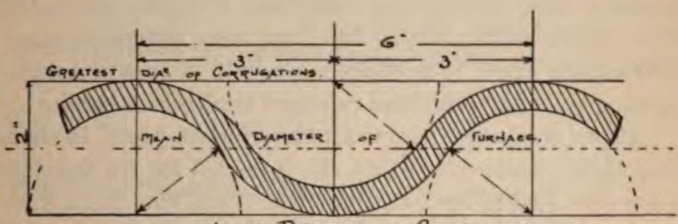
"In 1882 Fox invented his rolling-mill, which is still used by the Leeds Forge Company for the production of both Fox's corrugated and Morrison's suspension furnaces, the latter being an improved design lately introduced. These furnaces are made from Siemens-Martin steel ingots, rolled into plates of the required dimensions under ordinary plain rolls.

"Three sides of a plate are sheared, and on the fourth side the development of the saddle is marked and punched out

Test pieces are taken from the scrap thus produced and tested for tensile, elongation, and bending.

"The plate is then formed into a tube and lap-welded by water gas, the welding by this process being so efficient that in a series of tests lately made the tensile strength across the weld was found to be but slightly less than that of the original plate.

"This welded tube is next heated in a special furnace, and is then placed in Fox's patent mill, the rolls of which are corrugated; and in order to get the tube between the rolls, the top roll is worked out and in longitudinally by hydraulic power.



FOX'S CORRUGATED BOILER FURNACE.

"Complete corrugations are formed by one revolution, but a few turns are given for finishing, and the furnace is withdrawn as a perfectly cylindrical corrugated tube.

"Flanging is the next process, and although this has hitherto been done by hand, arrangements are being made for flanging by hydraulic pressure.

"This undoubtedly will be a great improvement, because of the fact that the greater the amount of work put on the back ends of furnaces of any type, the greater is the liability to crack, especially if the steel is of high tensile strength.

"The furnace is now complete, the final process being annealing.

"The furnace which is the greatest rival to the Fox is that known as 'Purves' ' patent, and is manufactured by Messrs. John Brown and Company, of Sheffield. This furnace was patented in 1880 by Mr. Purves, late of Lloyd's

Registry, and consists of a series of thickened ribs—9in. between the centres—the part between the strengthening ribs being of plain cylindrical form.

"A novelty in this furnace is the method of manufacture, as it is the first furnace of unequal section and the first furnace not made from an originally plain plate of equal thickness throughout.

"The 'Purves' flue is made from a Siemens-Martin ingot.

"Rectangular section slabs sufficient for two flues are formed from these ingots under a 15-ton hammer, the slabs being about 7¼ inches thick, and their length being equal approximately to the length of the flues required. Special roughing rolls convert the slab into a ribbed plate 1¼in. thick, which is then cut in two by a very powerful pair of shears, and, after re-heating, each half is passed through finishing rolls until the final required thickness is reached.

"At one side edge there is a piece of plain plate, 15 inches wide, to the centre of the first rib, intended for the front end of the flue, and at the other side the plain part is 23 inches wide and intended for the back end of the flue, and the thickness in both these side edges is increased by an ⅛in., to allow for the thinning which takes place during flanging, etc.

"After being sheared at the edges and slotted to the shape required, the plate is bent into a circular form by a special hydraulic press.

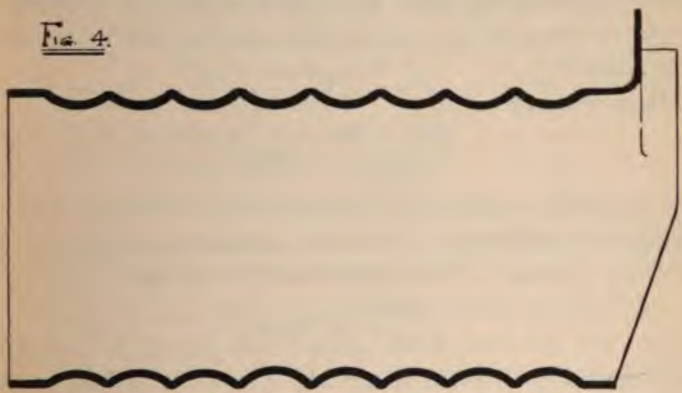
"The edges are then welded together by the insertion of glut-pieces, the plain parts being welded first and the ribs afterwards. Annealing is the next process, and after being withdrawn from the annealing furnace, the flue is converted into a perfectly circular tube by a very ingenious hydraulic press. It is then flanged in the ordinary way, and finally annealed again."

The latest design introduced is that known as Morrison's Suspension Furnace, which is a modification of both the Fox and Purves type, and a combination of the good features of each. It is manufactured at the same place, and in exactly the same way as the Fox—the same processes being employed throughout.

Another important feature in connection with such furnaces is that special provision is made for renewing them without interfering with front plates—and this is effected by cutting away the lower half of the flanged end where it connects to combustion chamber tube plate—so that it terminates in a plane inclined at about 120° to the horizontal, and consequently the furnace if required to be renewed can be raised, and tilted at the front end—thus enabling the flanged part at back to pass through the hole in front plate, without disturbing anything else. See Figure 4—which shows (in Morrison's Suspension Furnace) how this can be done.

During the five years from 1886 to 1891 a series of the most extensive and expensive experiments ever attempted

Fig. 4.



were carried out on various types of furnaces—under the supervision of the Board of Trade and Lloyd's Registry. The whole of the furnaces were full sized, and the conditions of test were of the most elaborate and stringent character. Nothing less than six furnaces had to be constructed for each design—the approximate length being 6ft. 6in., and the thickness ranging from $\frac{5}{16}$in., up to $\frac{9}{16}$in. They were all fitted —riveted in fact to properly constructed chambers, and tested to destruction by hydraulic pressure, in order to get at their actual strength, from which was deduced their re-

spective collapsing constants, and such constants are found in the following manner.

If we know the collapsing pressure of any corrugated furnace, it is very evident from what has been already said that if we multiply the diameter in inches by the collapsing pressure in lbs., and divide by the thickness, the result will give us the strength co-officient or constant, that is:

$$\frac{P \times D}{T} = C.$$

Where P = collapsing pressure in lbs. per square inch
D = diameter in inches
T = thickness in inches
C = constant for strength.

Take, for example, Fox's first steel corrugated furnace, whose diameter was 38in. and thickness $\frac{3}{8}$in., the collapsing pressure being 600 per sq. in. We want to find its collapsing constant?

Here we have

$$\frac{600 \times 38}{\cdot 375} = 60800$$

the required constant, and assume we are satisfied with a margin of safety of 5; then the constant for finding the working pressure of *this particular furnace* would be

$$\frac{60800}{5} = 12160$$

and the boiler pressure would be

$$\frac{C \times T}{D} = P.$$

In this case

$$\frac{12160 \times \cdot 375}{38} = 120\text{lbs. } WP,$$

which is one-fifth of the collapsing pressure.

It must, however, be remembered that in determining such a "constant," it must apply to different diameters and thickness; and as the strength of the steel varies from 22 to 2[illegible] and from 26 to 30 tons, and as the results obtained fro[illegible]

actual test do not *exactly* agree with the increase of thickness, or the higher tensile of the steel, caused probably by not being truly cylindrical, or by not being of uniform thickness—all these and other things have to be carefully considered before deciding upon a "constant," which shall approximately apply to furnaces of different dimensions. Hence it is that all such "constants" have only been determined after due consideration of the information obtained from experimental flues tested to destruction.

The number of furnaces, with claims to suitability for the present high pressure, are not confined to Fox or Purves, for the Board of Trade and Lloyds have conducted experiments on full sized tubes of eight different designs, and taking them in the order of seniority or age we have

1—Plain furnace	..	Figure 5
2—Adamson's rings	..	" 6
3—Fox's corrugated	..	" 7
4—Purves' ribbed	..	" 8
5—Holmes' ribbed	..	" 9
6—Morrison's suspension		" 10
7—Deighton's corrugated		" 11

Before describing the above, it is very desirable to make some reference to the experiments.

To us, in Australia, it does seem mighty strange that the Board of Trade and Lloyds, after carrying out such a splendid and exhaustive series of tests conducted under their own supervision, and in strict accordance with conditions framed by themselves, get such inharmonious results. One would naturally expect and look for standard rules, common to both, as the outcome of their experience. But it appears the two greatest authorities in the world, in treating the results of actual test, cannot even agree to measure the same furnace in the same way. Their constants and notation, not only for furnaces but for other parts, are different, and although as a whole, both sets of Rules are reliable and safe, still any one getting out a boiler to pass the Board of Trade and Lloyds has to make a double set of calculations—

Fig. 5.

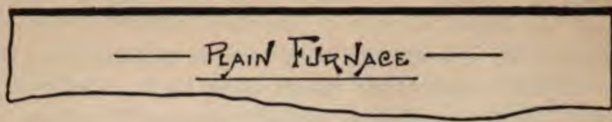


Fig. 6.

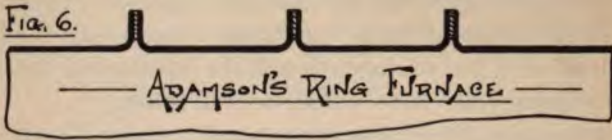


Fig. 7.

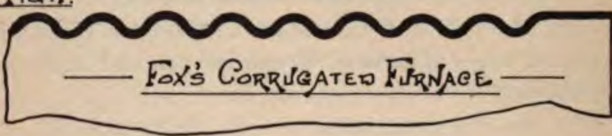


Fig. 8.

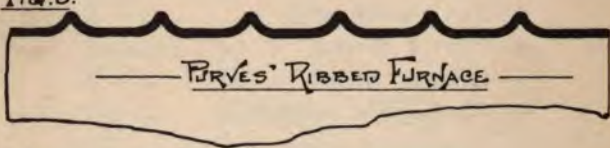


Fig. 9.

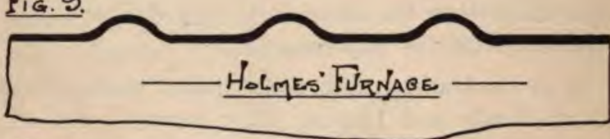


Fig. 10.

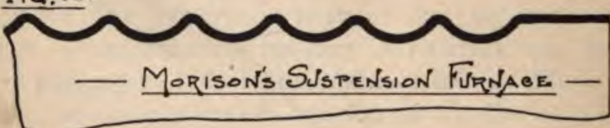
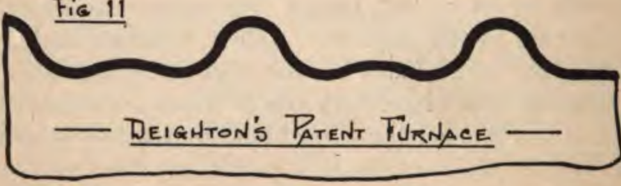


Fig 11



for in some things, what is good for Lloyds is not up to Board of Trade standard. The Board of Trade in measuring corrugated furnaces took the mean diameter—that is, the greatest + the least ÷ 2 = mean; lately, however, this has been altered, and the *least* external diameter is now taken, which must commend itself as being more sensible, while Lloyds take the greatest diameter over the corrugations.

Then in plain furnaces, the Board of Trade rule is

$$\frac{99000 \times \text{Thickness}''^2}{(L' + 1) \times D''} = \text{W.P.}$$

This L + 1 appears to be scarcely consistent—if we take two furnaces of similar diameter and thickness, only one is 5ft. and the other 8ft. long—in the one we add $\frac{1}{5}$th (20 per cent.), in the other an $\frac{1}{8}$th ($12\frac{1}{2}$ per cent.), which of course causes a corresponding, but unequal decrease in the working pressure.

Then again Lloyds, in furnaces of certain thickness, have the following

$$\frac{1000 \times (T - 2)}{D} = \text{W.P.}$$

Now this — 2 means that $\frac{2}{16}$th of an inch is deducted from the actual thickness of furnace plates *to allow for corrosion;* which seems to be questionable if not arbitrary, because now, when not neglected, corrosion is certainly the exception and not the rule.

In my early days I had to assist in doing a lot of general millwright's work, and remember well that almost every country blacksmith was a thorough conservative and had a screw tackle all of his own, and when a set of bolts and nuts by any mischance became mixed, the language used was more forcible than polite, but the idea of having a "thread" similar to his neighbour was out of the question all the same. In church matters, the same thing held; if one congregation stood up to sing and pray, another in duty bound sat down —they could not and would not agree to do the

same thing in the same way, and it certainly appears as if this doctrine has impregnated and permeated engineering science so far as boiler construction is concerned. To us, in Australia, it can scarcely be considered unreasonable, if we expect and hope that the results obtained from such valuable tests would be embodied in standard formulæ, having the same constants and notation, and which like Whitworth's thread would be of national application.

Referring to the figures, it will be seen that all the furnaces are of steel, because for present pressures iron, as we understand it, is seldom or never used.

The strength constants for each furnace have been increased from time to time, and this increase is partly due to the almost perfect method of construction; but lately the alteration has been caused principally by increasing the tensile strength of the steel.

When the first furnaces were made, the tensile of the steel was very low, not more than 22½ tons; but its strength has gradually risen, until in the most recent tests its tensile reached 29 tons, and the results proved two things—1st, that the strength of such furnaces increased approximately in proportion to their thickness; and 2nd, that the strength also increased in direct ratio as the tensile of the steel increased.

This, however, was under the cold water test, but it is very questionable and extremely doubtful if material of such high tensile is suitable for the very trying conditions, such as cleaning and banking fires, sweeping tubes, opening of doors, local heating, forced draught, and all variations of temperature with which it has to contend; besides, recent experience goes to prove (and especially in rigid furnaces) that this high tensile is a mistake, has already given trouble, and will probably end in failure.

Fox has constructed 30,000 corrugated furnaces, and the result of their extensive experience is that they advise steel of a lower tensile to be used, and not to exceed 27 tons

while Purves and others advocate as equally suitable steel of 28 to 30 tons, but it is suggestive that the German makers, Shulz and Knandt, of Essen, who have constructed 16,500 such furnaces, state very distinctly that 16,000 were made from steel whose tensile ranged from 23 to 26 tons, and that cracking, splitting, or trouble of any kind was unknown, but the other 500, which were made of high tensile steel, gave more trouble than the 16,000, so much so that the firm declines to take orders, unless the strength is reduced to an extent which, in their opinion, will make it reliable.

Purves has constructed 10,000 ribbed furnaces, which have done good service; and so far as the cold tests are concerned, Purves' and Fox's furnaces are of equal strength, and have the same constant, but, when under steam, Purves' is by far the more rigid of the two—in fact, it is simply a plain furnace with solid rings—and it has been clearly shown from actual test that as the strength and stiffness of the ribs, or rings, is a constant quantity, their strength decreases materially as the thickness of the furnace plates increase.

In the discussion on Morrison's paper, the question of high and low tensile was one of the principal points of contention, but the preponderance of evidence—opinion and experience—was that steel of such high tensile was too hard, and that the increased strength under the cold water test was only obtained at the expense of its ductility and reliability under steam.

One of the greatest advantages claimed for the corrugated over the plain surface is that it is sufficiently elastic longitudinally (and yet not too much so) to take up, provide for, and prevent any injury due to unequal expansion and contraction, caused by the well-known variation of temperature in different parts of the furnace.

The amount of elasticity has, however, until lately, been an unknown quantity, but the experiments made by the firm of Shulz and Knandt, of Essen, and carried out at great cost, on full-sized furnaces, to ascertain the longitudinal elasticity, are exceedingly valuable. In Morrison's paper,

all the details, illustrations, and diagrams are given in full; but the following will give a practical idea of the results.

The testing apparatus consisted of a hydraulic press, in which the furnace to be tested was placed horizontally between the body of the press and the hydraulic ram; these two representing the ends of a boiler. The load was applied longitudinally in increments of 15lbs., each increment on the pump representing 20·08 tons on the ram. On the completion of each increment, the shortening of the furnace was measured, the pressure taken off, and the length again measured in order to ascertain the permanent set, if any.

At intervals, the pressure was allowed to remain for a time, then released and again applied; but there was practically no difference in the results, if no permanent set had taken place.

The furnaces experimented on were—five of the corrugated type, one with (two) Adamson's rings and one Purves' ribbed. All these were made by Shulz and Knandt, with the exception of the Purves, which was manufactured by Brown and Co., of Sheffield.

The tensile of four of the corrugated tubes ranged from 21 to 25·4 tons per square inch. Adamson's rings were about the same, but one of the corrugated had a tensile of 29·2 tons. The tensile of the Purves' is not given, but was supposed to be from 26 to 30 tons.

All the furnaces were 70·86in. long, and the diameter outside corrugation ranged from 41·14in. to 41.45in., and the thickness ·393in., ·46in., ·472in., ·555in., ·393in., ·551in., and ·472in.

Adamson's rings were 37·4in. inside and 38·5in. outside the plain parts, while Purves' was 37·12in. inside and 38.07 in. outside.

For a comparison of results, a uniform pressure of 100 tons was applied to the end of furnace. In the corrugated the compressions were in the order of thickness given ·134in., ·112in., ·093in., and ·075in., per 36in. of length; in the Adamson ·021in., and in the Purves ·0071in. Further

the pressures required to shorten these furnaces $\frac{1}{32}$ in. (or ·01325in.) per 36in. in length would be for the given order of thickness 19·27, 24·74, 30·73, and 41 tons respectively. The other, whose corrugations were very shallow, required 92 tons.

Adamson's rings took 145·2 and Purves' ribbed furnace 300 tons to compress them the same amount—viz., $\frac{1}{32}$nd of an inch.

Such experiments show that even in the five corrugated furnaces the amount of elasticity is very small—less, perhaps, than was anticipated—but they also conclusively prove that the corrugated is by far the most elastic, that Adamson's rings possess a very fractional quantity, while Purves' has practically none at all; and this extreme rigidity is very much against its giving satisfaction with steel of a high tensile strength.

We don't want a furnace material that has to be nursed like a sick child, and where the opening of smokebox doors, cleaning fires, sweeping tubes, or banking fires is a source of anxiety to the engineer he cannot be everywhere and watch everybody; but we want furnaces made of stuff that will buckle and bend, that in case of over-heating will stretch and come right down on the fire bars without splitting or cracking, and after that to be examined, and, although out of shape, to be still sound.

Such a material has been put into many boilers, and can be easily supplied; the only difficulty is a competitive and commercial one, which has compelled makers to increase the tensile against their own experience, but anything short of the qualities just stated can never be thoroughly reliable.

We now proceed to consider the rules and formulæ which apply to the seven different designs, and also give practical examples by the Board of Trade and Lloyds' Rules.

The Board of Trade Rule for plain steel furnaces (Figure 5), when the material and workmanship are of the highest class, means that, if we want to use the highest constant—viz., 99000, the longitudinal seams must

be welded and thoroughly *annealed afterwards,* or the longitudinal joints either double riveted with single butt straps or single riveted with double butt straps, all rivet holes drilled in place after bending, the plates taken apart afterwards and the "burr" on the holes removed, and all the holes slightly countersunk from the outside, besides which the tests of the steel must also be satisfactory.

Rule—

$$\frac{99000 \times \text{Thickness in inches squared}}{(\text{length in feet} + 1) \times (\text{Diameter in inches})} = \text{WP per sq. in.}$$

EXAMPLE. A furnace 40in. external diameter, 7ft. long, and ½in. thick. Find its working pressure?

Here—

$$\frac{99000 \times \cdot 5^2}{(7 + 1) \times 40} = 77\text{lbs., the working pressure.}$$

Take the same furnace when the material and workmanship are of the very *worst* class, which means that the longitudinal seams are lapped, single riveted, and not bevelled. For such a case the constant is reduced from 99000 to 66000, which is exactly one-third, or 33 per cent. less, consequently the working pressure will be reduced in the same proportion —$77 \div 3 = 26$ and $77 - 26 = 51$lbs., working pressure, or

$$\frac{66000 \times \cdot 5^2}{(7 + 1) \times 40} = 51\text{lbs. WP.}$$

Besides the two constants used in present example, there are 13 others which apply to the various classes of work. Particulars and details will be found in the Board of Trade Rules.

Respecting the Rule just given, a little reflection will show it can only apply up to a certain point, after which it becomes absurd, and for two reasons:

1st.—Because we soon reach a thickness, beyond which it would not be prudent to go, as any addition would have a weakening instead of a strengthening effect, resulting in the fire side of the plates being burnt.

2nd.—The same holds with regard to the *number* of strengthening rings that can be fitted to any plain furnace.

The length is measured between the ring centres, and if we assume that up to 10 feet the strength is inversely as the length, then it is evident that a strengthening ring in the centre of a 10 feet furnace would make it twice as strong, because it would then be 5 feet long, and whatever working pressure the 10 feet was entitled to would be doubled; and if divided by rings into lengths of $2\frac{1}{2}$ feet between the centres, its strength and working pressure would be quadrupled *by the first Rule;* but if we multiply this last pressure by the diameter in in. and divide by the thickness in in., we will find that the steel would be subjected to a crushing strain, far exceeding what is allowed, and then the limiting formula would come in to prevent this.

To illustrate, and leaving out the + 1 for the sake of explanation, assume a furnace 10 feet long, 40in. in diameter, and $\frac{3}{8}$in. in thickness. The constant for *steel* being 99000, we have

$$\frac{99000 \times \cdot 375^2}{10 \times 40} = 34\text{lbs. the working pressure,}$$

and the crushing strain would be very small, viz.:

$$\frac{P \times D}{T}$$

In this case

$$\frac{34 \times 40}{\cdot 75} = 1800\text{lbs. per sq. in.,}$$

and if we fit a ring round its centre, the strength would be doubled; the working pressure would be 68lbs., and the crushing strain would be in the same ratio, viz.,

$$1800 \times 2 = 3600\text{lbs. per sq. in.}$$

So far we are all right, because for steel the amount allowed is 4950lbs. per square inch, which must not be exceeded, and here we are still well within it, being 1350lbs. under the 4950. If we fit two more rings, which would divide the furnace into four 2ft. 6in. lengths, then by the first rule the working pressure would be quadrupled, and we

would get $34 \times 4 = 136$lbs. *W.P.*, but the crushing strain would increase in the same ratio, and we would have

$$\frac{P \times D}{T} = \frac{136 \times 40}{\cdot 75} = 7253\text{lbs. per square inch}$$

as the crushing strain, which is far in excess of the limit allowed; therefore, in such a case the working pressure must be determined by the "limiting formula" which is for steel:

$$\frac{9900 \times \text{Thickness in inches}}{\text{Diameter in inches}} = \text{WP.}$$

Hence we have—

$$\frac{9900 \times \cdot 375\text{in.}}{40} = 92\text{lbs. WP,}$$

or, what is the same thing,

$$\frac{4950 \times (\cdot 375 \times 2)}{40} = 92\text{lbs. WP,}$$

showing clearly how the limiting formula checks and prevents all unnecessary increase in the number of strengthening rings, because no matter how many are fitted, they get no credit if the crushing strain exceeds 4950lbs. per square inch; in all such working cases the working pressure is determined by the second or limiting rule.

Let us now see how Lloyds' Rule applies to our 40in. furnace, 7ft. long and ½in. thick.

For *plain furnaces,* Lloyds have only two constants, and this is probably due to the fact that they will not pass any work that is not first class, which of course simplifies this particular item. For Lloyds, the strength of plain furnaces to resist the collapse is calculated from the following formula when the length of plain cylindrical part of the furnace exceeds 120 times the thickness of the plate—

$$\frac{1075200 \times T^2}{L \times D} = \text{working pressure per square inch}$$

Where T = thickness of plate in inches

„ D = outside diameter in inches

„ L = length of furnace in feet.

If strengthening rings are fitted the length between the ring centres is to be taken.

Applying this to our example, we have

$$\frac{1075200 \times \cdot 5^2}{7 \times 40} = 96\text{lbs., the working pressure,}$$

which is 4lbs. more than the Board of Trade.

For formula for short furnaces, see Rules.

Corrugated Furnaces.

Figure 7 represents Fox's corrugated furnace, in which the depth of corrugations is generally $1\frac{1}{2}$in., and the distance between the centres of corrugations 6in.

The Board of Trade, in measuring the furnace, take the outside diameter at the bottom of the corrugations.

The constant 14000 is derived from the actual experiments on full sized steel furnaces, which have been tested to destruction, and it provides a fair and reliable margin of safety in every case.

The rule is—

$$\frac{14000 \times T}{D} = \text{Working pressure per sq. in.}$$

Where C = 14000, the strength constant,
,, T = thickness in inches
,, D = outside diameter in inches, measured at bottom of corrugations.

Applying this to a corrugated furnace, whose diameter outside corrugations is 42in., the thickness being $\frac{1}{2}$in., we have

$$\frac{14000 \times \cdot 5}{42} = 166\text{lbs. working pressure,}$$

which would be allowed for steel whose tensile was not less than 26 and not more than 30 tons.

For Fox's or Morrison's.

By Lloyds, the strength of steel corrugated furnaces whose tensile is from 26 to 30 tons, the corrugations being 6in. apart and $1\frac{1}{2}$in. deep, is calculated from the following:—

$$\frac{1259 \times (T - 2)}{D} = \text{Working pressure.}$$

Where T = thickness in sixteenths of an inch
D = outside diameter of corrugations in inches.

Taking the same furnace as we did for Board of Trade Rule, the outside diameter of corrugations would be 45in.

Therefore we have $\dfrac{1259 \times (8 - 2)}{45} = 167$lbs. working pressure.

In this particular case there is practically no difference between the Board of Trade and Lloyds, as the working pressures come out the same; but if we take a furnace of 2ft. 6in. inside corrugations, $\frac{3}{8}$in. thick, and calculate the W.P. by both rules, we would get results differing materially, for the Board of Trade would give 170lbs., whereas Lloyds would only allow 148lbs.—a difference of 22lbs. in the W.P.

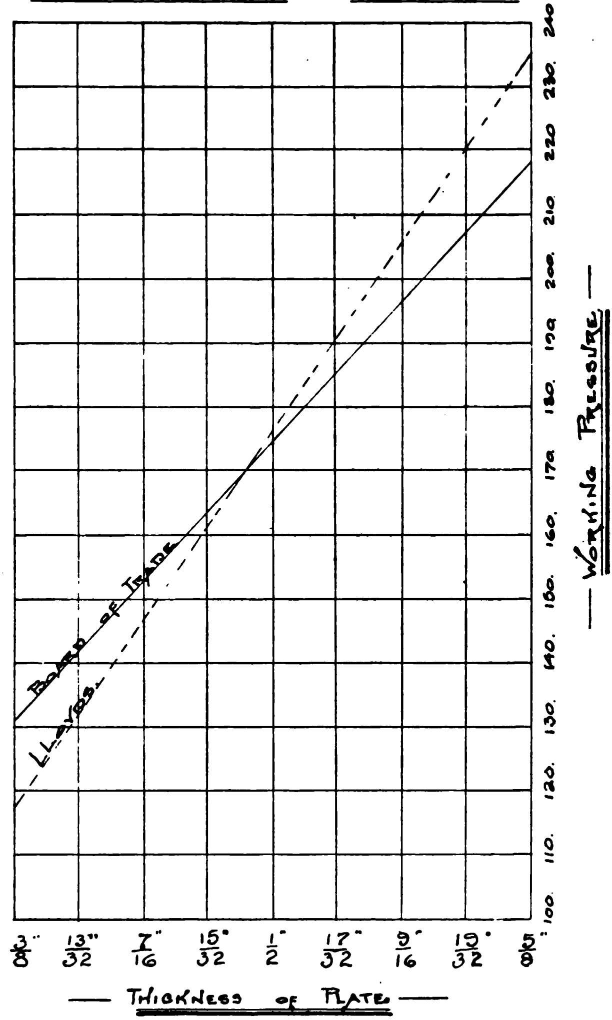
Again, if we take the same 2ft. 6in. furnace when it is $\frac{5}{8}$in. thick, and make the same calculation, we will find it comes out the other way; for Lloyds would give 294lbs., but the Board of Trade only 280lbs.—a difference of 14lbs.

The diagram Figure 12 shows the peculiar variations in the working pressures, and it will be noticed that the thinner the plate the greater the difference, which difference lessens as the thickness increases, until at $\frac{1}{2}$in. it disappears, and any increase over the $\frac{1}{2}$in. brings about the very opposite result, for in a 2ft. 6in. furnace, $\frac{3}{8}$in. thick, the Board of Trade allow 22lbs. more than Lloyds; but if the same furnace be $\frac{5}{8}$in. thick, then Lloyds would allow 14lbs. more than the Board of Trade.

These variations are due principally to the theory of inevitable corrosion, as previously explained by the introduc-

Fig 12.

— Corrugated Furnace — 3'·6" Diameter —



tion of (T — 2), the effect of which is more severe on thin than on thick plates.

That such a difference—or, in fact, any difference—should exist is to be regretted, more especially in such grand experiments, where there was nothing emperical or assumptive, so far as the basis or foundation they had to work from was concerned; but engineers, like doctors, differ, and there is no doubt that both Lloyds and the Board of Trade are quite prepared, from their own point of view, to support and defend their action, but to practical minds of ordinary perception, it is questionable if the soundness of the logic used would be very apparent or of a convincing character; while the respective explanations might, or might not, be conducive to increasing their dignity and influence.

Figure 8 represents the Purves-Brown steel ribbed furnace, the rib centres being 9in.

The Board of Trade Rule for this furnace is

$$\frac{14000 \times T}{D}$$

so that in constant and notation it is exactly the same as Fox's.

Lloyds' Rule for the Purves steel-ribbed furnace is *not* the same as for Fox's. The constant for Fox's is 1259, but for Purves' it is

$$\frac{1160 \times (T - 2)}{D} = W.P.,$$

and for a furnace 44in. external diameter over plain part and ½in. thick, we would get

$$\frac{1160 \times (8 - 2)}{44} = 158\text{lbs.},$$

or 1lb. less than the Board of Trade.

Comparing the Fox, Purves, and Morrison furnaces, it may be noticed that in the Fox the inner corrugations, which are always next the fire, if not kept clean are apt to crack and give trouble, that the scale or deposit is more difficult to remove, and that a certain percentage of the width of fire-

grate is sacrificed, whereas in the Purves the ribs or rings are entirely clear of the fire, the plain surface is more easily cleaned and more likely to keep clean, and that the effective fire-grate is larger for the same diameter of furnace; while, on the other hand, those undoubted advantages are discounted by the Purves being of unequal section and exceptionally rigid, and the fact that the strength imparted by the ribs decreases considerably as the thickness of the plain part increases.

Morrison's claims to be a combination of the two, embodying all the good and leaving out the bad points which practical experience has shown to be necessary, and he describes it as follows:—"The features retained from the Fox designs are: the disposition of the material in a form which gives the greatest resistance to collapse, uniform thickness and uniform strength throughout, distribution of strains uniformly throughout the length of the furnace.

The features retained from the Purves flue are: the strengthening formations are in the water space, and are protected from the fire; there are no narrow cavities for the accumulation of scale, and there are equal facilities for scaling and cleaning."

From the results obtained by testing to destruction six full-sized furnaces on his principle, and also from experience in actual working, Morrison does not appear to claim too much. It is without doubt a great improvement on Fox and Purves, and is rapidly coming into general use.

Figure 9 represents the Holmes' furnace, in which the corrugations are not more than 16in. centres, and not less than two inches high.

The Board of Trade gives no rule for this furnace, but Lloyds' experiments on full-sized furnaces of this design resulted in the following:—

$$\frac{945 \times (T - 2)}{D} = \text{Working pressure.}$$

Where T = thickness in sixteenths of an inch
D = outside diameter of plain part in inches.

Comparing this with Lloyds' latest constant allowed for the Fox and Morrison—viz., 1259—we have

$$\frac{945}{1259} = \cdot 75$$

which means that the Holmes' furnace of the same tensile, diameter, and thickness is 25 per cent. weaker, which practically bars its adoption.

As in Holmes', the Board of Trade give no rule, but Lloyds' experiments bring it out almost the same as Holmes', thus—

$$\frac{912 \times (T - 2)}{D} = \text{Working pressure.}$$

Figure 10 represents Morrison's suspension furnace, and it will be noticed that the curves are easy, but strong, that the strengthening ribs or wings are well clear of the fire, that it is of uniform thickness throughout and easily cleaned.

Figure 11 represents Deighton's furnace, and it will be noticed that all the curves are easy and strong. This furnace has the good points of the ribbed furnace, with the advantage of being of uniform thickness.

The above is a brief description of seven furnaces of different design, and all with claims to suitability for the present high pressures, but the competition is practically confined to four, viz., Fox, Purves, Morrison and Deighton's, and in the present state of our knowledge Morrison and Deighton appear to have effected a good combination, for they comply with and provide for the conditions of practical working much more than the others, and in that sense their furnaces commend themselves as being the most suitable and reliable under steam.

Temperature of Furnace Plates.

The question of "How thick furnace plates can be safely used without the fire side of the plate being burnt" has been

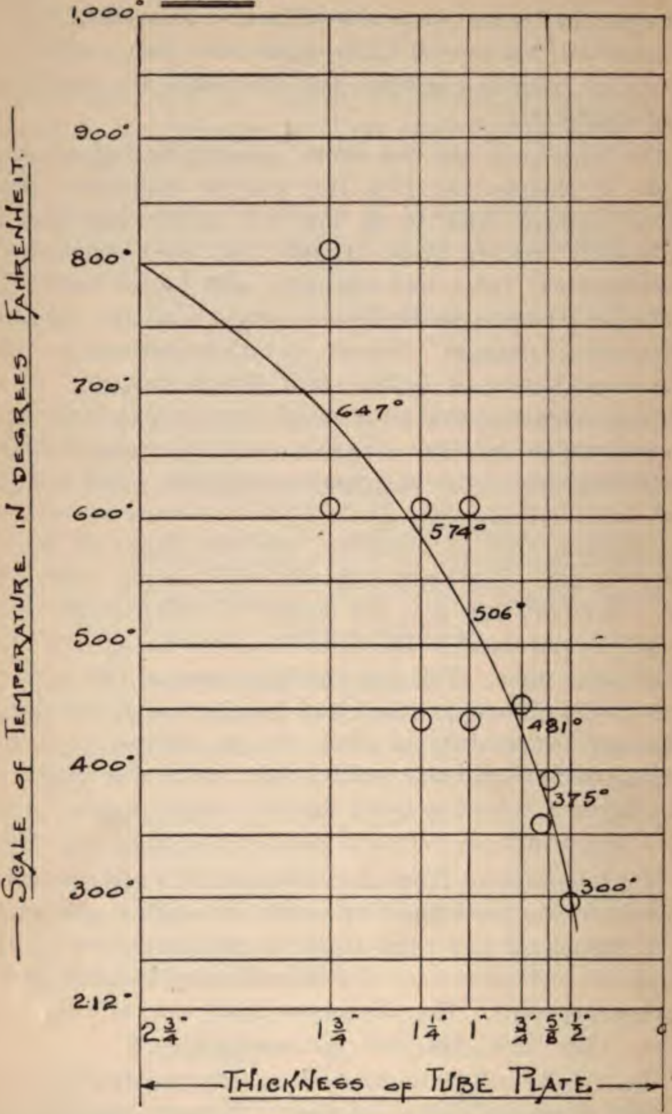
often asked, but up to the present never satisfactorily answered, neither has the exact difference in temperature between the fire and water side of furnace plates been clearly determined, but several experiments have been carried out lately by different engineers, and the results are interesting and suggestive.

Dr. Kirk made the first effort, a description of which is given in *Engineering,* 15th July and 9th September, 1892. What induced Kirk to go into the matter was that in Admiralty boilers great trouble had been experienced through leaky tubes, and especially with forced draught, a difficulty which in the merchant service is almost unknown with natural draught. Yarrow, in his admirable paper "On the construction of boilers with forced draught," very sensibly suggested that with forced draught leaky tube ends in combustion chambers were to a considerable extent caused by fitting tube plates of excessive thickness, which allowed tube ends and fire side of tube plates to become overheated. To confirm this, or otherwise, was the object of Kirk's experiments. His apparatus was very simple, consisting of a wrought iron pot, the bottom of which at the start was $2\frac{3}{4}$in. thick, and in the centre of which was fitted a $2\frac{1}{2}$in. steel boiler tube. This pot was quite open at the top, and the water, of course, boiled and remained at a practically constant temperature of 212° It was placed, or rather built, over an ordinary smith's fire, which was blown by two tuyeres, the effect being that the bottom was subjected to a temperature in excess of that due to a forced draught.

The temperature difference between the fire and water side of bottom was ascertained by means of fusible plugs, which were hammered into holes drilled on the fire side, while the thickness of bottom was reduced by turning so much off for each experiment. The thicknesses tried were as follows:—$2\frac{3}{4}$in., $1\frac{3}{4}$in., $1\frac{1}{4}$in., 1in., $\frac{3}{4}$in., $\frac{5}{8}$in., and $\frac{1}{2}$in.

The results will be understood by referring to Figure 13, which represents Kirk's diagram. The base line gives the thickness of tube plate at each experiment, while the vertical

— Fig. 13.—


1,000°
900°
800°
700°
600°
500°
400°
300°
212°
647°
574°
506°
481°
375°
300°
2 3/4"
1 3/4"
1 1/4"
1"
3/4"
5/8"
1/2"
0
— Scale of Temperature in Degrees Fahrenheit. —
Thickness of Tube Plate

lines are the corresponding temperatures of the tube end on the fire side, the water side in every case being at 212°. The highest and lowest possible temperatures are marked by small circles, and the curve drawn is the mean.

Where the thickness is greatest, the spots are a long way apart; but when we come to practical thickness ($\frac{3}{4}$in., $\frac{5}{8}$in., and $\frac{1}{2}$in.), the curve and the spots are reasonably close. But such results cannot be taken as conclusive, because they are made under conditions widely different from actual working; and in determining the difference of temperature by means of fusible plugs, such results must always be uncertain. What they do show is that the thicker the plate the greater the temperature difference between the fire and water sides, and the results suggest that $\frac{3}{4}$in. is the maximum thickness for tube plates, and that if they were reduced to $\frac{1}{2}$in. we should have very little trouble with leaky tubes, provided the internal surfaces were thoroughly clean.

Mr. Blechynden made some experiments somewhat similar to those described, but his results differ very materially from Kirk's, the difference in temperature between the fire and water sides of plates being much less, the discrepancies showing that this important point is still in its experimental stage, and that we must wait for more definite information.

What concerns us most is to realize the absolute necessity of keeping the internal surfaces clean, and to remember that a very thin coating of oil or grease will allow plates, and especially in furnaces, to become overheated. In support of this it may be desirable to refer to a particular experiment of Professor Lewis's, in which he burnt a hole with the assistance of a blow-pipe through a small iron pot filled with water, but which had a layer of oily deposit $\frac{1}{32}$in. thick on the water side, whilst another similar pot with no deposit was entirely unaffected.

Again, Mr. Morrison has made some more recent experiments of a practical character, which he describes as follows:—

"To determine the temperature of furnace plates, he

placed a vessel 18in. in diameter over a specially made furnace, the temperature of which was 2000°. The bottom of the vessel was a steel plate $\frac{3}{4}$in. thick, and the temperature of this plate $\frac{1}{4}$in. from the fire side was 290° with a clean plate; but it was 400° with $\frac{1}{64}$in. of clean, hard scale, and when this scale was painted over with a coating of cylinder oil, making the thickness $\frac{1}{32}$in., the temperature of the plate rose to 650°."

These results have been confirmed in a paper by A. J. Durston, Engineer-in-Chief H.M. Navy, and read before the Institution of Naval Architects.

"In an open vessel having a bottom $\frac{1}{4}$in. thick, placed over a furnace having a temperature of 2200°, the temperature of the fire side of the plate when clean was 280°; but with $\frac{1}{16}$ in. of greasy scale it increased to 550°. In another experiment with a closed vessel $\frac{5}{8}$in. thick and 110lbs. steam pressure, the temperature on the fire side, with clean surfaces and clean water, was 430°; but with $\frac{1}{16}$ in. greasy deposit it rose to 550°, and using more grease the temperature registered was 617°."

Such results, although not conclusive, are valuable, and should impress upon all the importance of purifying the feed water and extracting the deposit dirt and grease before discharging it into the water space. It has taken a long time to realise the absolute necessity of doing this, but of late mechanical filters for this special purpose, which are coming into general use, are turning out to be fairly effective, and no doubt such appliances will be considerably improved.

Reflecting upon the above, practical questions naturally present themselves, and the mechanical engineer would likely say to himself, "Well, if with such a thin coating of oil or grease the temperature difference between the fire and water sides is so great, at what temperature would such plates become overheated? And what about the internal strains set up through the particles of the steel being so much hotter on the one side than on the other? Under such conditions the furnace plates would be very much in the same

state as a high-pressure piston-rod that got blazing hot on one side only, and which through unequal expansion would probably bend,'' etc.

Definite answers to such inquiries are as yet not forthcoming. But what has been done must and will form a strong incentive towards recognising the supreme importance of keeping the internal surfaces of high-pressure boilers, and especially furnace plates and parts exposed to the impact of flame, perfectly clean, for then we know that the temperature difference is reduced to a minimum.

Respecting the effect of temperature on mild steel when exposed to tension, the exhaustive experiments carried out by the United States Government are instructive.

Briefly expressed, it was found that, from 32° up to about 300°, the strength decreased, but, above that, increased until its maximum was reached at from 400° to 600°, beyond which its strength decreased rapidly. Also that its elastic limit, from 32° to 600°, decreased throughout; at zero it was 35000lbs., while at 600° it was only 20000lbs. per square inch.

With thin plates (from $\frac{1}{2}$in. to $\frac{3}{4}$in.) and 180 lbs. of steam pressure, the temperature approaches the brittle temperature of steel. Hence the necessity of employing a material of low tensile, because the experiments went to show that the lower carbon steels retain their maximum strength and ductility over a greater range of temperature than those of high tensile strength. But it should be remembered that although there may be material difference between the fire and water side of furnace plates, still the *mean* temperature throughout the entire section of the plate will be considerably less, although what that mean is at present is more or less assumptive.

Red-Hot Furnace Crowns.

What is the proper thing to do if by any mischance the furnace crowns become overheated?

To such a question at least ninety per cent. of those responsible for or in charge of boilers would probably

answer—"Ease the safety valves and draw fires; but on no account or under any circumstances is the cold 'feed' water to be discharged or come in contact with the overheated plates, because in such a case the amount of steam that would be instantaneously generated would be so great that the whole structure would explode."

This is the "bogey"—the "Will o' the Wisp" theory, that teachers and text books have drilled into us for so many years, and many of our best men, such as Scott, Russell, Maudsley, and Field, Parker, and even Professor Tyndall himself, have endorsed it. In almost all inquiries into *mysterious* boiler explosions the scientific evidence invariably tended towards supporting this doctrine—usually expressed somewhat as follows:—"That although in some cases and under ordinary conditions the simple force of the steam might rend a boiler, still that alone would not explain or account for the lifting and projection of a boiler from its seat, or for one half being hurled 100 yards, while the other remained in its original position, etc.; also, that such explosions are due to a force momentary in its nature, tearing the boiler plates at the instant of its generation, and before there was time for its transmission to the safety valves, and that the existence of a force of this nature, sudden in its action, instantaneous in its duration, had been firmly established in many minds, and that a force different in form and greater than the simple pressure of the steam was the principal agent. Therefore the argument was that the introduction of cold water on red hot plates meant the instantaneous generation of steam of such volume and pressure that no number of safety valves could prevent the bursting of the boiler," etc., etc.

That such a doctrine has been received, accepted, and adopted without question is probably due to the difficulty of disproving it, which could only be done by actual tests under steam. Such tests, according to the above notions, would be not only costly but difficult and dangerous. A little common-sense reflection, assisted by increased knowledge and experience, generated and developed by a spirit of scepticism,

resulted in the belief that what had been laid down as gospel regarding this sudden mysterious force was, after all, mere assertion, expression of opinion, and assumption, without the slightest tangible logic or evidence to support such statements.

The credit of exposing this fallacy is entirely due to the spirited enterprise of the Manchester Steam Users' Association, who, under great difficulties and at great expense, carried out a series of experiments, resulting, not in the explosion of the boiler, but in the complete explosion of the theory: because they proved that the showering of cold water on red-hot furnace crowns did not, and cannot, generate this mysterious sudden and uncontrollable force. Therefore this "bogey" is, to all intents and purposes, dead and buried.

The experiments were carried out under the supervision of Lavington Fletcher, the well-known engineer, and his report, giving a most graphic description and all details, will be found to be most interesting and instructive reading. (See *Engineering,* January, February and March, 1891.)

The tests were carried out on an ordinary Lancashire boiler, with the special object of ascertaining the effect of showering cold water on red-hot furnace crowns.

Boiler was 27 feet long, 7 feet in diameter, having two 36in. plain furnaces running through it, and without any Galloway tubes. Both shell and furnaces were $\frac{7}{16}$in. thick, the total grate surface in both furnaces being about 36 square feet. It had the usual fittings and two safety valves, one 3in. and the other 4in. in diameter. The feed pipes were placed about 6in. above the crowns, and were carried along the whole length of fire bars, about 6ft. 6in., and arranged in such a manner (being perforated with holes) as to discharge the cold water directly on the exposed surface of the red-hot plates. The amount of water capable of being pumped as feed was $5\frac{1}{2}$ cubic feet per minute, a quantity sufficient for at least four boilers of this class, under ordinary working conditions, as they were anxious to make the tests under the worst possible circumstances.

The first experiment consisted in blowing the water down to a point 16 inches below the crowns, one of the safety valves being seated and the other propped open, with 6lbs. of steam on the guage. When the plates became red hot the cold feed was turned full on, and the pressure rose from 6 to 12lbs. in 11¼ minutes.

This experiment was repeated with both safety valves seated, and in ¾ of a minute the pressure rose from 6 to 27lbs., but afterwards gradually fell to 6lbs. This was with a good clear fire, 7in. thick, and damper full open, the feed being discharged full bore, at the rate of 342 gallons per minute. The safety valves were loaded to 50lbs. per sq. in., but even under such extraordinary conditions the steam only rose 21lbs., a force totally insufficient to even lift the safety valve.

Several other experiments followed, but instead of blowing the water out it was allowed to evaporate by shutting off the feed. In every case the subsequent experiments more than confirmed the first, for when the feed was turned on the increase was very fractional, and in some the pressure actually fell.

Generally the experiments clearly proved that it is impossible for any large quantity of steam to be suddenly generated by showering cold water on red-hot plates, under even such severe and exceptional conditions, and far more so under the ordinary method of introducing the feed.

Although such experiments effectually dispose of the theory so long and so persistently taught, the question still arises as to whether it would be wise or prudent in all cases to turn the cold feed on when a boiler is short of water. The answer to such a question requires both consideration and caution.

As the experiments showed that when both safety valves were open the introduction of cold feed resulted in a fall instead of a rise in pressure, we might reasonably argue that in some cases where shortness of water takes place, with the *engines running*, a man would be quite justified in turning on the cold feed, *provided he in every such case*

eased his safety valves first, for then the slight increase of pressure due to the cold water on the overheated plates would scarcely affect the steam gauge, but would be discharged through the safety valves. Under such circumstances turning on the feed would be the best thing to do, and would very probably bring the water to its proper level, cool the plates, and prevent any explosion, besides giving the fireman more time and more confidence in drawing fires.

It, however, becomes a very different matter if shortness of water takes place when the engines are standing, because the experiments demonstrated (but under most exceptional conditions) that the showering of cold feed on red-hot plates raised the steam from 6 to 27lbs. in $\frac{3}{4}$ of a minute, and it is quite possible that this sudden extra pressure, added to that on the steam gauge, might just turn the scale on the already overheated and weakened plates, and bring the crowns down.

On the other hand, it must be remembered that in the particular experiment where the pressure rose 21lbs. both safety valves were closed, and I am inclined to think that even under such circumstances there would be comparatively little danger in turning on the cold feed, provided the engineer or fireman *thoroughly* eased the safety valve *first.*

This view is suggested and supported by the fact that, in the experimental boiler, the feed was purposely arranged so as to discharge direct on the whole of the exposed surface, and in such quantity and direction as could not possibly occur in ordinary working; and if, as previously stated, the safety valves were well eased before opening the feed valves, it is very probable there would be very little, if any, increase of pressure.

In summing up his able report, Fletcher is evidently of opinion that, in the majority of cases, turning on the cold feed would be the best thing to do; but he very properly qualifies that opinion by pointing out "that the results of the experiments, though of great value, are not conclusive enough to justify a hard and fast rule being laid down,

certainly not one that could be absolutely adopted under all circumstances without consideration or discretion."

In Lancashire boilers and in others of similar design to the experimental boiler described—that is, where the flues are plain, made in continuous rings, lap-jointed, single-riveted, of good material, with all holes drilled in place, etc. —the rules for finding the working pressure on such furnaces differ considerably from those used in marine boilers, and there is also a material variation in the value of the constants used. To give such rules or compare them would only result in confusion: hence, the desirability of selecting one thoroughly reliable rule, and which has been deduced from modern practice.

Longridge's Rule for plain long and short lap-riveted flues is about the best, being recognised as a standard, and is as follows:—

$$\frac{C \times T^2}{D \times \sqrt{L}} = P.$$

Where P = working pressure in lbs. per sq. in.
 " T = thickness in sixteenths of an inch,
 " D = diameter (outside) in inches,
 " L = length in feet,
 " C = 200, the constant.

Example.—Take a furnace 36in. outside diameter, 25 feet long, and ½in. thick. At what pressure would we work it?

Here we have $\frac{200 \times 8^2}{36 \times \sqrt{25}} =$ 71lbs. working pressure.

If this furnace were 16 feet long, instead of 25,

we would have $\frac{200 \times 8^2}{36 \times \sqrt{16}} =$ 90lbs. working pressure;

and if it were 12ft. 6in. long,

we would have $\frac{200 \times 8^2}{36 \times \sqrt{12{\cdot}5}} =$ 101lbs. working pressure;

which shows clearly in what ratio (according to modern practice) the strength decreases as the length increases.

Strength of a Sphere.

What is the difference between a cube and a sphere of the same diameter?

If we take a cube whose diameter is unity, or one (such as one inch or one foot), put it into the lathe and turn it into a sphere, when finished it will be the same diameter as the cube, but not the same capacity or volume. How much have we turned off? Nearly one half. The material in the cube was 1, but the amount left is only ·5236; so that the difference between a cube and a sphere of the same *diameter* is as 1 to ·5236. Therefore, if we want to find the capacity of any ball or sphere, we simply cube the diameter and multiply by ·5236: for example a 12in. cube would contain $12 \times 12 \times 12 = 1728$ cubic inches; but a sphere of the same diameter would only contain $12 \times 12 \times 12 \times \cdot 5236 = 904{\cdot}78$ cubic inches.

Spherical vessels are the best of all forms for resisting internal pressure, and also for containing the greatest volume within a given amount of surface, from which it follows that the steam pressure always has a tendency to make any containing surface assume a spherical form.

If we investigate the internal force tending to burst a sphere we will find by similar reasoning employed in finding the bursting strain of shells that it would rupture through the longest line we could draw through it, which, of course, would be its diameter, and therefore the area of its diameter in square inches × the pressure in lbs. per square inch would = total force of the steam.

The "formula" is exactly the same as that used for transverse strain in shells, viz.: (same notation).

$$\frac{P \times D^2 \times \pi}{4} = \text{total force of the steam,}$$

or what is more usual and the same thing—

Area in sq. in. × pressure = total force of the steam.

Take an example: A spherical ended iron steam dome is 60in. internal diameter, and the steam pressure is 80lbs. per sq. in. Find the force tending to burst it?

By the first we have

$$\frac{80 \times 60^2 \times 3{\cdot}1416}{4} = 226195\text{lbs. the force.}$$

equal in this case to 100 tons.

By the second we get the same: $80 \times 60^2 \times {\cdot}7854 =$ 226195lbs. the force.

This force is resisted by the tensile strength of the iron, and its amount is equal to the circumference of the sphere × its thickness, and when the force of the steam and the strength of the iron are equal we have the same expression as for transverse strain in shells, viz.:

$$\frac{P \times D^2 \times \pi}{4} = \pi \times T \times S$$

$$\text{And } T = \frac{P \times D}{4 \times S}$$

Therefore a sphere is twice as strong as a cylinder of the same diameter and thickness is longitudinally.

The relative strengths of a sphere and a cylinder will be clearly seen from the following comparison:—

Take a sphere whose diameter is unity, or one, and whose circumference will, of course, be 3·1416, its area being of necessity ·7854. Take a cylinder of the same diameter and sectional area; it must be ·7854 long.

Taking the material in the cylinder, we have the sum of the two sides, viz., $\cdot7854 \times 2 = 1{\cdot}5078$; but the material in the sphere is $3.1416 \times 1 = 3{\cdot}1416$, just twice as much.

From what has been said it is plain that in setting out steam domes or cambered ends, if we have a radius equal to the boiler barrel, the ends will be as strong as the shell longitudinally, and the camber will form portion of a circle whose diameter is double that of the boiler shell.

Flat Surfaces.

We now come to consider how flat surfaces should be treated, as seen in the steam and water spaces, combustion chambers, flat ends, etc.

The theoretical investigations of scientific men in connection with the strength and stiffness of plain flat surfaces are largely assumptive, besides being extremely complicated, and the "formula" given by Rankine and others for the strength of a flat, circular plate supported all round the edge (that is, throughout its entire circumference) is never used in practice.

When flat surfaces are under steam pressure they are subjected to bending and buckling strains, and to ensure them retaining their original form we support them with stays. Fire boxes and water spaces are usually secured by stays screwed into both plates, and either riveted over with a bat-head or fitted with nuts on one or both sides; while steam spaces are supported by long iron or steel rods having screwed ends, with nuts and washers or double plates on one side or both. Sometimes, but not often, stays forged with T palms riveted between double angle-irons are fitted; while some prefer gusset stays riveted in double shear to shell and end plates.

Steam and water space staying is always (where practicable) set off in squares, and in all such, each stay has to bear a strain equal to the area supported in square inches multiplied by the boiler pressure in lbs.—that is, the pitch^2 × boiler pressure = strain on each stay. If the steam spaces are stayed by gussets or diagonal stays, then each stay has to bear a stress equal to the pressure exerted on a sector of the circular area, and the resultant tension is greater than on a direct longitudinal stay, because the strain in the former has to be transmitted at an angle while in the latter it is straight and direct. If, therefore, we fit diagonal stays, we must remember and apply the following rule:—

"The tension on a diagonal stay is equal to the tension which a stay placed perpendicular to the flat surface would

sustain, divided by the cosine of the angle which the diagonal stay makes with a perpendicular to the supported surface—or, to put it shortly, the surface supported in sq. in. × pressure in lbs. ÷ cosine of the angle, equals the strain on diagonal stay. For example, assume we have to support 90 square inches, with 100lbs. of steam per sq. in., then the strain would be 9000lbs. for a longitudinal stay; but if instead we fix a diagonal bolt-stay at an angle of 60°, we must divide the 9000lbs. by the cosine of this angle, and as the cosine of 60° is ·5, we have 9000 ÷ ·5 = 18000, just double the strain of a direct stay."

(The cosine of an angle is found by dividing the base by the hypothenuse, and must, of course, be always less than 1.)

The rule for gusset stays is different, and more simply expressed. It is as follows:—

$$\frac{\text{Area supported in sq. in.} \times \text{boiler pressure}}{\text{Depth of web at narrowest part} \times \text{thickness of web in in.}}$$

= working stress per sq. in.

Example.—The working stress on a gusset stay is not to exceed 7000. It has to support 420 square inches of surface, and the boiler pressure is 100lbs. If we make it ½in. thick and 12in. deep at the narrowest part of web, will it be strong enough?

Here $\frac{420 \times 100}{12 \times \cdot 5} = 7000$lbs., so that this stay complies with the requirements.

In proportioning the staying of flat surfaces it is usual to neglect the rigidity of the plates to be stayed, and it is always assumed that each stay sustains the pressure on the area due to the square of the pitch, and when well fitted this is the actual strain; but in steam spaces, where portable stays are fixed with pin or other joints, they cannot be depended upon for an equal distribution of strain.

Such stays should never be used unless absolutely necessary, and, when they are, special care should be taken to see that the single and double eyes are truly bored out in the

machine, and the pins turned with as much care and accuracy as a piece of engine work, thereby ensuring that each one will do its fair share of the work.

The flat surfaces between the stays when under pressure are considered to be in the same condition as a continuous girder or beam, uniformly loaded, fixed at the ends, and supported at the points of attachment of the stays. The tendency of the pressure, if the material is deficient in strength or stiffness is to split up the plate between the stays.

According to the laws which govern beams (and which will be referred to further on), the strength of flat plates to resist bending is directly proportional to the square of the thickness, and inversely as the square of the pitch—that is, if the stays are pitched 4in. each one has to support 16 square inches; but if we double the pitch, and make the pitch 8in., then each stay has to sustain the pressure exerted on 64 square inches (just four times as much), so that in the latter case each stay must be made four times stronger than in that having a 4in. pitch.

The same holds good for the plate's thickness, but in a different way, and for a different reason. If we want to find the working pressure for a plate $\frac{1}{2}$in. thick, where the stays are pitched 10in. centres, and if the strength constant (deduced from actual experiment) is 100 for screw stays with bat-heads, then we have the rule (Lloyds') : —

$$\frac{C \times T^2}{P^2} = \text{greatest working stress allowed per sq. in. on stay}$$

Where C = 100,
,, T = thickness in 16ths,
,, P = pitch in inches:

$$\text{Therefore } \frac{100 \times 8^2}{10^2} = \frac{6400}{100} = 64\text{lbs. W.P.}$$

If, however, we double the plate thickness and make it 1in. thick, we would have $\dfrac{100 \times 16^2}{10^2} = \dfrac{25600}{100} = 256$lbs. working pressure, four times greater than when the plate was $\frac{1}{2}$in. thick.

By doubling the thickness we not only double the amount of material, but we also double the resisting leverage of the plate, and that is the reason why the strength and stiffness varies directly as the square of the thickness;— or, what is the same thing, if we consider it as a beam fixed at both ends and uniformly loaded, we might say that its strength varies directly as the square of the depth of the beam, which in this case is 1in.

When the $\dfrac{\text{pressure} \times \text{pitch}^2}{2}$ = strength of the iron × thickness^2, then the strain and the strength are equal, and the slightest increase of pressure would rupture the plate; but we always arrange the material so that it is at least five times stronger than the strength it has to bear, so that the strength is to the strain as 5 is to 1.

The rules which govern the working pressure allowed on flat surfaces have to a very great extent been deduced from actual experiments carried out on full-sized plates and stays, every care being taken to have all the conditions exactly the same as in actual working, excepting, of course, the impact of heat and flame, etc. The results obtained are exceedingly valuable, thoroughly reliable, and the strength "constants" can be used with all confidence.

The Board of Trade has at least ten (10) different "strength constants," to suit the different classes of work, and they range (for iron) from 192 down to 36, so that the working pressure must vary in the above proportion. For steel plates the number is about the same, the highest being 240 and the lowest 39·6.

The Board of Trade formula for flat surfaces is as follows:—

$$\frac{C \times (T + 1)^2}{S - 6} = \text{working pressure.}$$

Where T = thickness of plates in 16ths.
S = surface supported in square inches.
C = "constant" as per class of work.

Example.—If iron stays are pitched 9in. and the plates are $\frac{1}{2}$in. thick, find the working pressure when the plates are not exposed to the impact of heat or flame, and the stays fitted with nuts only? Referring to the Board of Trade Rules we find that the "constant" for such conditions is 90. Therefore we have

$$\frac{90 \times (8+1)^2}{81 - 6} = \frac{7290}{75} = 97\text{lbs. W.P.}$$

Had the plates been of steel the "constant" is increased from 90 to 112·5, so that the working pressure would be greater in proportion.

Lloyds' Rules for flat surfaces embrace about the same number of "constants" for iron as steel, but both formula and notation are different, which is to be regretted, as there is no material difference in the working pressure allowed. Lloyds' Rule is of a more practical nature than the Board of Trade's. At all events, it commends itself as being more easily understood, and consequently will be better appreciated. It is as follows:—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per sq. in.}$$

Where T = thickness of plate in 16ths
P = greatest pitch in inches
C = "constant" as per class of work.

It is somewhat difficult to compare the respective values of the Rules given for flat surfaces by the Board of Trade and Lloyds, because the conditions governing the "constants" are certainly not in harmony; but taking the same example —9in. pitch and $\frac{1}{2}$in. plates, with the stays screwed and fitted with nuts only—we find the "strength constant" is 120. Therefore

$$\frac{120 \times 8^2}{81} = \frac{120 \times 64}{81} = 95\text{lbs. per sq. in.,}$$

the working pressure, a difference of only 2lbs.

Respecting direct iron stays for flat surfaces, the maxi-

mum stress allowed on iron per square inch of sectional area is 7000lbs., and 9000lbs. for steel; but only under certain conditions. If the stays are welded or worked in the fire, or if the ends are jumped up, the allowance is reduced to 5000lbs. per square inch of net section, while the diameter of all such stays is always measured from the bottom of the thread.

The size of stays, etc., is found by the following:—

$$\frac{\text{Surface in sq. in.} \times \text{pressure in sq. in.}}{\text{Stress in lbs. per sq. in.}} = \text{area of stay in sq. in.}$$

$$\frac{\sqrt{\text{Area in sq. in.}}}{\cdot 7854} = \text{diameter in in. at bottom of thread,}$$

$$\frac{\text{Stress in lbs. per sq. in.} \times \text{area}}{\text{Pressure in sq. in.}} = \text{surface supported in sq. in.}$$

$$\frac{\text{Surface supported in sq. in.} \times \text{pressure}}{\text{Area in sq. in.}} = \text{stress in lbs.,}$$

$$\frac{\text{Stress in lbs. per sq. in.} \times \text{area in sq. in.}}{\text{Surface supported}} = \text{boiler pressure.}$$

When a stay is attached by bolts or rivets, the total area of such bolts or rivets should exceed the area of the stay, and when the bolts and rivets are in single shear their area should not be less than 20 per cent. more than the stay; but if in double shear their collective area may be 25 per cent. less than the stay, and if the stay is welded the 20 per cent. should be reduced to 10, and the 25 per cent. increased to 30 per cent.

Generally all stays should be well fitted and carefully tightened, the workman making a special point of satisfying himself that every stay takes its fair share of the strain. When diagonal stays are fitted the palms should always be forged out of the solid, all holes drilled, bolts turned, and the rivet or bolt area in excess of the stay. If gussets are fitted, their sectional area should be considerably in excess of that required for diagonal stays, as their design is weaker, the greatest strain being on the bottom edge.

All gussets should be of exceptionally good material, and the rivets always in double shear; the collective rivet area 20 per cent. more than in a diagonal stay in single shear, but 25 per cent. less than the area required for a diagonal stay if in double shear. Sometimes gussets are fitted rough off the punching or shearing machine. This should always be planed off, to insure the removal of any injury to the material through punching or shearing.

As a fair approximation, the ultimate strength or breaking strain of stays which will give a good elongation and contraction of area may be set down as follows:—

Copper stays = 15 tons, or 33600lbs. per sq. in. of sectional area
Iron stays = 23 tons, or 51520lbs. per sq. in. of sectional area
Steel stays = 28 tons, or 62720lbs. per sq. in. of sectional area.

Another point to which special attention should be directed is the mistaken idea that in water spaces if the stays are screwed into both plates, and riveted over with a bat-head, it is as good as if the stays were fitted with nuts. A little reflection will show the fallacy of this, because if the stays are, say, $1\frac{1}{4}$in. in diameter and the plate $\frac{3}{8}$in. thick, it practically amounts to making $1\frac{1}{4}$in. bolts with nuts $\frac{3}{8}$in. thick; whereas to make the two proportional the depth of the nut should be equal to the diameter of the stay. In single-ended boilers it sometimes happens, but not often, that in staying the back combustion chamber plates to the back end plate nuts are only required on the fire side, as the end plates are so much thicker, but with the present pressure this is seldom or never done. In all such staying it is not necessary to make the depth of the nut equal to the diameter of the bolt, because the thickness of the screwed plate practically represents so much thickness of nut—that is, if the stays are $1\frac{1}{4}$in. and the plate $\frac{3}{8}$in., the nuts may be $1\frac{1}{4}$ less $\frac{3}{8}$, which would make them $\frac{7}{8}$ thick.

Respecting the screw threads, the holding power of the stay is reduced in proportion to the depth of the thread. The angle of Whitworth's thread is 55°, with $\frac{1}{8}$th rounded

off top and bottom, and the following table gives the reduction in diameter due to the depth of threads:—

12	threads per inch reduces diameter ..	·1066
11	„ „ „ „ ..	·1163
10	„ „ „ „ ..	·128
9	„ „ „ „ ..	·1422
8	„ „ „ „ ..	·16
7	„ „ „ „ ..	·1828
6	„ „ „ „ ..	·2133
5	„ „ „ „ ..	·256
$4\frac{1}{2}$	„ „ „ „ ..	·2844
4	„ „ „ „ ..	·32
$3\frac{1}{2}$	„ „ „ „ ..	·3657
$3\frac{1}{4}$	„ „ „ „ ..	·4037
3	„ „ „ „ ..	·4266
$2\frac{7}{8}$	„ „ „ „ ..	·4452
$2\frac{3}{4}$	„ „ „ „ ..	·4654
$2\frac{5}{8}$	„ „ „ „ ..	·4875
$2\frac{1}{2}$	„ „ „ „ ..	·512
2	„ „ „ „ ..	·64

From the above will be seen the necessity of always measuring screwed bolts and stays at the bottom of the thread. For instance, take a 3in. steam space stay; screwed 4 threads per inch, the diameter would be reduced ·32in., making it 2·68in. The area of a 3in. bar is 7 square inches, and, if of steel, its breaking strain would be $7 \times 63000 = 441000$lbs. The area of a 2·68in. bar $= 5{\cdot}6$ square inches, the breaking strain of which is $5{\cdot}6 \times 63000 = 352800$, a reduction in strength of 88200lbs.

To find the amount of holding surface in the screwed end of a stay bolt we measure across the bottom of thread, to which we add half the depth of thread, which gives us the mean diameter. Then $3{\cdot}1416 \times$ mean diameter $\times$ number of threads in nut $\times$ depth of thread gives the total holding surface in square inches.

Assuming depth of nut and diameter of bolt to be the

same, and the screw a good fit, then, screwed end, nut, and body of stay are well proportioned for their work.

To show the material increase of strength by fitting nuts instead of bat-heads, it is desirable to refer to some valuable experiments made under the supervision of the Board of Trade for the special purpose of ascertaining the actual resistance of steel plates to water pressure when supported by stays, as in the flat surface of boilers. The stays were in some cases screwed through both plates and riveted over, and in others screwed through both plates and fitted with nuts on both sides.

The thickness of the experimental plates were $\frac{5}{16}$ in, $\frac{7}{16}$ in., and $\frac{9}{16}$ in., and all the stays were screwed 10 threads to the inch. Experimental boxes were constructed for each of the above thicknesses; all plates and stays of steel. They (the boxes) were $56\frac{1}{2}$in. square from centre to centre of rivets, and in each were fitted 16 stays set off in squares, and pitched $11\frac{5}{16}$ in. centres.

This number of stays was selected in order to ensure that, except as regards the influence of heat, the test should be as nearly as possible in accordance with ordinary working conditions. Hitherto such experiments have been carried out with a comparatively small number of stays, where it was found that a certain amount of support was given to the plate by the sides of the box giving a higher pressure than in the combustion chamber of a boiler, where the plates are supported by a large number of stays. In this case of the 16 stays, the 4 central ones were certainly quite free from any side influence, and the experiments proved them to have been more severely strained than any of the others. Without going into the details of the tests, it will be sufficient for the present purpose to take the $\frac{7}{16}$ in. plates, as fitted with bats and nuts. As before stated, the pitch was $11\frac{5}{16}$ in., and the stays $1\frac{1}{2}$in. diameter. In the box with bat-heads, when the pressure got up to 250lbs. per square inch, the riveted bats showed signs of giving way in the central space (that is, the four stays furthest away from the edge), and

at 280lbs. the bats flew off and the stays drew through the plate. In another box exactly similar, excepting that the stays were fitted with nuts, a very different result was obtained. The pressure was raised to 700lbs. per square inch, when the box failed by the plate cracking suddenly in four places with a sharp report.

The average results of the experiments may be condensed and put as follows:—

$\frac{5}{16}$in. plates, with bat heads, gave way at 185lbs. per sq. in.
$\frac{7}{16}$in. " " " " " 280lbs. "
$\frac{9}{16}$in. " " " " " 450lbs. "

while the increased strength for nuts was for

$\frac{5}{16}$in. plates, fitted with nuts, gave way at 450lbs. per sq. in
$\frac{7}{16}$in. " " " " " 700lbs. "
$\frac{9}{16}$in. " " " " " 980lbs. "

The relative bursting pressures might be expressed thus:

($\frac{5}{16}$in. plates) $\frac{450}{185} = 2{\cdot}4$ times stronger with nuts than with bats.

($\frac{7}{16}$in. plates) $\frac{700}{280} = 2{\cdot}5$ times stronger with nuts than with bats.

($\frac{9}{16}$in. plates) $\frac{980}{450} = 2{\cdot}2$ times stronger with nuts than with bats.

These figures prove conclusively the large increase of resisting power due to the fitting of nuts. It should, however, be noted that in the experimental boxes where the stays were fitted with nuts the stays were larger and the nuts deeper than would be required in practice, the object being to ensure that the plates must give way first, so that their actual strength and stiffness could be ascertained. The bulging of the plates was accurately measured, and although the actual bursting pressures are high, still it was proved that considerable stretching at the stay holes took place long before the plate cracked. It is, therefore, very evident that, in deciding upon the working pressure for such flat

surfaces, it is necessary to consider at what pressure the stay holes begin to stretch, and in the rules given ample provision is made for this.

Strength of Bolts and Nuts.

A very natural question in connection with this subject would be this: Suppose we screw the end of a 1in. bolt, does the cutting of the thread affect its strength?—that is, would it be as strong as a plain bolt of the same sectional area? The experimental data on this point is rather limited, but the following is both interesting and instructive.

Mr. Brunel tested the tensile strength of screwed bolts and nuts made of Shropshire iron, the bolts ranging from $\frac{5}{8}$in. to $1\frac{1}{4}$in. in diameter. The stress was applied between the head and the nut, the distance between them being 16in.

The length of the screwed part was $3\frac{1}{4}$in. In most of the tests the bolt snapped at the bottom at the screwed part. To find to what extent the screwing of a bolt or stay reduces its tensile strength, four $1\frac{1}{4}$in. bolts and nuts were tested, on which the screwed part was enlarged to $1\frac{1}{2}$in. in diameter. All the bolts gave way in the shank, the average breaking weight being 25·2 tons per square inch, showing an increased strength of 2·2 tons per square inch as compared with the plain parallel $1\frac{1}{4}$in. screwed end, which would have 7 threads per inch. From the tests we may infer that the strength of the $1\frac{1}{4}$in. bolts is reduced 2·2 tons, or 8 per cent., by screwing. The following gives the results of the experiments:—

Diameter of Bolts in inches.	Breaking Weight in tons.	Breaking Weight in tons per sq. inch.
$\frac{5}{8}$	$10\frac{1}{4}$	32
$\frac{3}{4}$	12	27
$\frac{7}{8}$	$15\frac{3}{4}$	25
1	20	25
$1\frac{1}{8}$	21	21
$1\frac{1}{4}$	29	23

Respecting the nuts, those 1in. deep (or $\frac{8}{10}$ths of the diameter) stood well; when $\frac{7}{8}$in. deep (or $\frac{7}{10}$ths of the diameter) the thread stripped; and when $\frac{3}{4}$in. deep (or $\frac{6}{10}$ths of the diameter) thread also stripped. For ordinary practice a depth of nut $\frac{4}{5}$ths of the diameter has been found to be sufficient; but to make sure and allow for contingencies the depth should not be less than the diameter of the bolt; and, in the case of cylinder covers, etc., where the nuts are often unscrewed, they should be even deeper than the diameter of the studs or bolts.

Combustion Chamber Girders.

In all, or nearly all, modern boilers the tops of combustion chambers are quite flat, and require to be supported in proportion to the exposed flat surface, and in so far as the pitch of stays and thickness of plate in chamber tops are concerned, the treatment is exactly the same as in the water spaces, with this difference, that the steam pressure which each stay has to sustain is transmitted to the girder, which on account of a number of practical considerations (explained further on) has to be constructed from 8 to 10 times stronger than the calculated stress it has to bear.

The "formula" for combustion chamber girders, as given in text books, gives no information which would satisfy the practical mind, and as it is both desirable and necessary that both engineer and boilermaker should know something of the why and wherefore, the following remarks on beams are given, being entirely confined, however, to those of rectangular section.

The word "girder" is almost always adopted by engineers as the name for beams which are supported at both ends and subjected to transverse strain, and the term "cantilever" for beams supported at one end only, and which have also to bear a transverse strain.

Every mechanic knows that any bar of iron or beam rectangular in section is stronger and stiffer if placed on edge, and will bear a much heavier load without bending

than the same bar or beam placed on the flat; also, that a given quantity of material made into, say, a hollow column would be much stronger to resist compression, buckling, or bending than the same amount of material whose sectional area was solid.

The laws which govern beams, and especially those of varied and irregular section, are complex, and require considerable mathematical knowledge which is quite beyond the scope of the present effort; but there are certain elementary principles and laws which, with the aid of experimental data, may be illustrated and described in such a way as to give a practical grip of what beams have to do, more especially those on combustion chamber tops.

In considering the loading of girders there are certain things we must know. We must understand the mechanical effect produced under the varying conditions of support; we must have a clear idea of the different stresses brought to bear upon the girder, and also in what manner the particular form of section affects the resistances to such stresses; we must also realise and provide for the different and distinct strains, and knowing the different strengths no material will be wasted in getting the maximum of efficiency.

For our purpose it is best to rely upon actual experiments of such men as Barlow, Fairbairn, Hodgkinson, Kirkaldy, and others, because in many cases scientific deductions have not been in harmony with practical results.

As in the present case we have to deal with beams of rectangular section only, their relative strength, in so far as it depends on the manner in which they may be loaded and supported, can be expressed in simple figures.

Let us consider six different methods of loading:—

1st. A rectangular beam of uniform section is fixed at one end and loaded at the other. How do we get at the total force tending to break or bend it?

The formula for this is simply expressed, viz.—

Weight in lbs. $\times$ length in inches $=$ total force or *bending moment,* as it is usually called, and is generally expressed in inch pounds.

Example.—Assume such a beam to be 40in. long, firmly fixed at one end, and that the total load or weight on the other end is one ton (2240lbs.), we want to find the total force tending to break or bend it,—or, as expressed in text books, we want to find its bending moment.

Here we have weight $\times$ length $=$ B.M.

In figures, 2240lbs. $\times$ 40in. $=$ 89600 inch pounds, the B.M.; and this method of loading gives the maximum B.M.

2nd. If in the same beam the same weight were distributed instead of acting on the end of a 40in. lever, what would be the difference in the bending moment?

In this case the weight $\times \dfrac{\text{length}}{2} =$ B.M.

$\therefore 2240 \times \dfrac{40}{2} = 44800$lbs., exactly one-half, showing No. 2 to be twice as strong as No. 1.

3rd. If the beam, instead of being fixed, rested on two supports, 40in. apart, the weight being in the centre and both ends loose and free, how would the B.M. be affected?

Under such conditions, we would have $\dfrac{\text{weight}}{2} \times \dfrac{\text{length}}{2}$ $=$ B.M.

In figures, $\dfrac{2240}{2} \times \dfrac{40}{2} = 22400$lbs.—the B.M., which is four times stronger than No. 1, and twice as strong as No. 2.

4th. Beam resting on the two supports, 40in. between their centres, but the weight uniformly distributed. How much would the bending moment be reduced?

Here we would have $\dfrac{\text{weight}}{2} \times \dfrac{\text{length}}{4} =$ B.M.

$\therefore$ the figures would be $\dfrac{2240}{2} \times \dfrac{40}{4} = 11200$lbs., 8 times stronger than No. 1, 4 times stronger than No. 2, and twice as strong as No. 3.

5th. The beam fixed at both ends, with the weight in the centre. How does its B.M. compare with the others?

Rule for this is: $\dfrac{\text{Weight} \times \text{length}}{8} = \text{B.M.}$

$\therefore$ the figures give $\dfrac{2240 \times 40}{8} = 11200$lbs., the same as in No. 4.

6th. Beam fixed at both ends and uniformly loaded. How much is the strength increased? or how much is the B.M. decreased?

Here we have $\dfrac{\text{weight} \times \text{length}}{16} = \text{B.M.}$

$\therefore$ our figures give $\dfrac{2240 \times 40}{16} = 5600$lbs., which is 16 times stronger than the first, 8 times stronger than the second, 4 times stronger than the third, and twice as strong as the fourth and fifth.

Summing up, we might put those examples in the following form:—

	Inch lbs.	Tons.
1st. When fixed at one end and loaded at the other, the total force or bending moment is	= 89600	= 40
2nd. When fixed at one end, but the load or weight evenly distributed ..	= 44800	= 20
3rd. Loose and free at both ends and loaded at the centre	= 22400	= 10
4th. Loose and free at both ends, but weight uniformly distributed ..	= 11200	= 5
5th. Fixed at both ends, with weight in the centre	= 11200	= 5
6th. Fixed at both ends and uniformly loaded	= 5600	= $2\frac{1}{2}$

The reason why the strength of the girder is so materially affected by the different methods of loading is that it is, and practically acts as, a *bent* lever, the long arm being measured from the point of support to the centre of the weight, the short or bent arm being always equal in length to the depth of the girder. Take No. 1, a rectangular beam of uniform section fixed at one end and loaded at the other. Assume the length to be four feet and the depth one foot, and that a ton weight is hanging on the end. Then it is evident that the

$$\frac{\text{Weight} \times \text{long arm}}{\text{short arm}} = \frac{1 \text{ ton} \times 4 \text{ feet}}{1 \text{ foot}} = 4 \text{ tons},$$

which, of course, represents the weight or stress the beam has to bear at the point of support. If the same beam, instead of having the weight at the end, had it equally distributed over the entire length, the stress on the point of support would be reduced one half, because the centre of gravity of the weight would be only half the distance it was from the point of support when the weight was on the end—that is, the mean weight of the equally distributed load would pass through a point exactly in the centre of the beam's length, which consequently lessens the leverage 50 per cent. and reduces the stress on the point of support from four to two tons. Again, if the same gorder were supported at both ends and loaded in the centre (which would represent a combustion chamber stay with one bolt), it can be shown that it would take four tons to break it, just four times as much as the same beam fixed at one end with the same weight on the other.

The weight in any girder supported at the ends and loaded in the centre is transmitted with proportional force to each point of support, which reacts on the girder with an intensity exactly proportional to the leverage, and if in this beam the weight be placed 1 foot from one support the distance to the other will be $\frac{3}{4}$ths of the lever's total strength, and whatever the weight may be the stress on the support

1 foot from the weight will be $\frac{3}{4}$ths of the total weight on the beam—that is, if the total stress were 4 tons one end would bear 1 and the other 3 tons. If the weight be in the centre, then the reaction would be equal, and each support would take 2 tons, just half the weight.

It can also be shown that in a similar girder supported at both ends, but uniformly loaded, it would take 8 tons to break it, as the mean leverage would be found to pass through a point half-way between the centre of the girder and the points of support.

In like manner, if the same beam were firmly and rigidly fixed at both ends, and uniformly loaded, it can be shown that the leverage is reduced and its strength increased in the proportion of 16 to 1—that is, it would be 16 times stronger than the same beam fixed at one end and loaded at the other.

It must be remembered, however, that the above deductions are all theoretical. The weight of the beam itself is not taken into account, while the material and workmanship is assumed to be perfect. In combsution chamber girders the weight of beam is of no practical moment, but in large structures, such as bridges, or in cast iron beams, having considerable span, the weight of the beam itself may represent the largest percentage of the load it has to carry, while in almost all cases (boilers not excepted) considerable allowance and provision has to be made for imperfect material and workmanship.

Combustion chamber girders, as usually fitted, are in the condition of beams supported at the ends, but variably or unequally loaded, and their strength is affected to a great extent by the manner in which the material is arranged to take the strain. On this important point there is a wide difference between theory and practice, but, as before stated, the results of actual experiment are quite sufficient for our present purpose.

Consider a girder at work—that is, the steam pressure on top plates taken by the stay bolts and transmitted to the

girder: the tendency is to bend under the load, and a little reflection will make it clear that the top fibres or particles must be in compression and the bottom particles in tension, and that the top and bottom edges are more severely strained than any other part; also, that this compression and tension affect the material in a gradually decreasing ratio as the distance of the affected particles from the top and bottom of the girder increases, until a point is reached where there is neither compression or tension. Assuming the material to be equal in tensile and compressive strength, this neutral surface, or neutral axis, as it is called, would be represented by a line drawn through the centre of the rectangular section, where the particles of the iron or steel throughout the length of the girder practically do no work, and are entirely unaffected by the stresses on the top and bottom.

It should, therefore, be carefully noted that the work done by the girder is very unequally distributed, that a large proportion of the section does no work at all, and that the strains upon the material decrease from the top until the neutral surface or axis is reached, when they vanish altogether; also, that the particles nearest the neutral line contribute least towards supporting the load; but, on the other hand, the farther away from this line the greater is the percentage of strain they bear in supporting the load, and therefore the neutral surface is the place where holes may be drilled and attachments made without affecting the strength of the girder. In all girders the exact position of the neutral line depends entirely upon the nature and arrangement of the material, and if the tensional and compressive powers are equal then this line will be in the middle of its depth, but not otherwise. If the beam be made of cast iron it would never do to make it rectangular in section, because this metal in compression is good for 48 tons per square inch, while its tensile strength is only 8 tons; therefore we must distribute the material in proportion to the respective strengths, and the depth of the

material above the neutral surface will be to that beneath it as the square root of 1 to the square root of 6, or approximately as 1 is to $2\frac{1}{2}$, or to make even figures, as 2 is to 5.

With steel, however, we may take the tension and compression as being nearly equal, which, of course, in the present case, simplifies the calculation, and in combustion chamber girders of uniform rectangular section the line of neutral surface will practically pass through the centre of the depth.

The total stress, or load, which the particles of the material in each half of the beam has to bear is found by multiplying the area of half the beam by the mean strain upon the fibres, while the total strain upon either half of the beam can be determined by multiplying the mean strain by the breadth and by half the depth.

To find the mean strain upon the material means finding a point where if all the pressure or load upon the beam were concentrated it would have precisely the same effect as under the actual working conditions, and this point is called in technical language the "centre of gyration" of the beam.

To explain this more fully: When work is done its amount is measured by multiplying the weight by its velocity. For instance, if we lift 33000lbs. one foot high in one minute we develop a certain quantity of work equivalent or equal to one-horse power; but if we take two minutes to lift the same weight, then the work done is only equal to half a horse power. If we take two engines having same sized cylinders, pressure, and revolutions, but with the stroke 2 feet in one to 4 feet in the other, then the work done by the long stroke is double that of the short stroke, because the same weight passes through double the distance in the same time, or we might say the same weight is raised to double the height, or double the weight to the same height in the same time.

If, therefore, we consider a fly-wheel of an engine in motion, and that this wheel has a broad rim, it is very evident that the material at and about the outside of rim

is moving with a greater velocity than the material at or near the inside rim; consequently the particles in the outer rim are doing more work than those on the inside, because being the same weight they pass through a greater space or distance in the same time. Therefore, if we want to get at the power stored up in this wheel, and because the parts of any revolving body must move at different speeds, it becomes necessary to find what the diameter of the circle would be in which the mean motion may be supposed to take place, or to find that point in the diameter where, if all the weight were concentrated, the dynamical effect would be precisely the same as under the ordinary running conditions. Such a point would be the "circle of gyration" of the fly-wheel, and its distance from the centre of motion would be the "radius of gyration." For convenience it is usual to take the mean diameter of the fly-wheel rim as the radius of gyration, as this is approximately correct, or, at all events, near enough for all practical purposes, and it is also usual to neglect the weight of the arms.

Coming back to the combustion chamber girder, the same principle applies. Like the fly-wheel, the weight, or load, to be supported, the work to be done, is most unequally distributed throughout the section. The particles nearest top and bottom edges take the lion's share of the load; the material at the neutral line does nothing; while a large percentage of the sectional area of girder is practically useless, so far as taking any strain is concerned.

Although the girder does not revolve, still it is just as necessary to find its centre and radius of gyration as in the fly-wheel, because the point that would give a true *mean* of all the unequal strains is the most important factor, without which we could arrive at no definite result.

Generally it may be accepted that the strength of solid rectangular combustion chamber girders varies inversely as the length, directly as the breadth, and directly as the square of the depth. It is important that this should be clearly explained.

1st. "Inversely as the length" means that by doubling the length we double the leverage, and although the resisting power of the section remains the same, still if 100lbs. on the end of a beam 20in. long would break it, then 50lbs. on the end of a beam 40in. long would have the same effect.

2nd. "Directly as the breadth." If we double the breadth we double the number of resisting particles, but we do not alter the leverage; consequently twice the breadth is simply twice the strength, but no more.

3rd. "Directly as the square of the depth" means that if we double the depth we not only double the amount of material, but we also *halve the leverage,* and consequently have four times the resisting power; hence it is that in beams of rectangular section the strength varies as the *square* of the depth.

If the section of a combustion chamber girder were square instead of rectangular the conditions and variations would be inversely as the length and directly as the *cube* of the side of the square, because in doubling the side of any square we get four times the material and twice the leverage, which, of course, makes it eight times stronger than the original beam, or, as is usually expressed, the strength of square beams is as the cube of their sides. The same applies to solid cylindrical beams, whose strength varies inversely as the length and directly as the cube of the diameter, for the same reason as that given for square beams.

No reference has been made to the difference between the strength and stiffness of beams, nor is it necessary for our purpose. The foregoing remarks on beams are essentially of an elementary character, but they may create an appetite for more information, and such information will be found in Anderson's "Strength of Materials," to which I am indebted. It is one of our best text books of science, and can be purchased from any bookseller.

Taking the sure ground of experimental facts, the following rule may be expressed:—

In beams of uniform sound rectangular section the

"bending moment," or total resisting power of all such beams in inch pounds is invariably equal to the area of section in sq. in. $\times$ radius of gyration $\times$ strength of material per sq. in.

The radius of gyration of a beam 1in. deep has been conclusively proved to be equal to $\cdot 2886$, which is the constant used to find the true resisting power of the section, so that the following rule would give the same result:—Depth^2 $\times$ breadth $\times$ $\cdot 2886$ $\times$ strength of the iron or steel per sq. in. This is also equivalent or equal to the resisting power of the girder.

In any rectangular beam, if we divide the square of the depth by 12 *and extract the square root the result is the radius of gyration.*

$$\left(\sqrt{\frac{1^2}{12}} = \cdot 2886\right)$$

The following example will perhaps make this clear:—Take a beam of steel 4in. deep 1in. thick and 20in. between the supports.

$$\text{Then } \sqrt{\frac{4^2}{12}} = 1 \cdot 154, \text{ the radius of gyration.}$$

Taking the strength of the steel at 61000lbs., we have

$$4 \times 1 \cdot 154 \times 61000 = \frac{\text{weight}}{2} \times \frac{\text{length}}{2}$$

$$\text{weight} = \frac{4 \times 1 \cdot 154 \times 61000 \times 4}{20} = 56315\text{lbs., the breaking}$$

load

$$\text{and } \frac{56315 \times 20}{4} = 281575\text{lbs., the bending moment.}$$

Having found bending moment, to find depth of beam we have

$$\sqrt{\frac{281575}{1 \times \cdot 2881 \times 61000}} = 4\text{in., the depth of beam.}$$

To find the stress on the material by the bent lever principle, we have

$$\frac{\frac{56315}{2} \times \frac{20}{2}}{1 \cdot 154} = 244000\text{lbs., the total stress;}$$

And as we have 4 square inches, then $\frac{244000}{4} = 61000$lbs., the stress per square inch.

It should always be remembered that the two rules previously mentioned mean one and the same thing—that is, the radius of gyration $\times$ depth $= 1 \cdot 154 \times 4 = 4 \cdot 616$, and depth$^2 \times \cdot 2886 = 4^2 \times \cdot 2886 = 4 \cdot 616$ also, so that we can use either, but it is more handy to square the depth and multiply by the constant $\cdot 2886$.

If the material were equally strained we should have no centre or radius of gyration, and the short or bent arm of the lever would be equal in length to the lever's depth; but because the strains are unequal the resisting power of the girder is reduced in the proportion of the square of its depth multiplied by its radius of gyration; in fact, the radius of gyration practically represents the short end of the lever, which is the foundation of this and other calculations where the stresses developed on the material are unequal.

The practical application of the above will be better understood and appreciated from the following examples, in which the strains on combustion chamber girders, fitted with 1, 2, 3, and 4 bolts, are worked out in figures, and are shown in a graphical form. Before working them out it will be necessary to give the Board of Trade "formula" for such girders, from which by comparison we can see what factor of safety is considered desirable, and also get a better idea why it is necessary to have different constants for girders having 1, 2, 3, or 4 bolts.

Girders for Flat Surfaces (Steel).—Board of Trade Rule.

When the tops of combustion chambers or other parts of a boiler are supported by solid rectangular girders the following formula is used for finding the working pressure to be allowed on the girders, assuming they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers, that the ends are fitted to the edges of the tube plate and the back plate of the combustion box, or in a double-ended boiler, the ends fitted to both tube plates:—

$$\frac{C \times d^2 \times T}{(W - P) \times D \times L} = \text{working pressure.}$$

Where W = width of combination box in inches
P = pitch of supporting bolts in inches
D = distance of girders, centre to centre, in inches
L = length of girder in feet
d = depth of girder in inches
T = thickness of girder in inches
C = 660 when girder is fitted with one bolt
C = 990 when girder is fitted with two or three bolts
C = 1100 when girder is fitted with four bolts.

The working pressure for the supporting bolts and for the plate between them is the same as given for flat surfaces and stays.

Referring to Figure 14, we have:—

Width of combustion chamber, 16in.

Girder, 4⅝in. deep and 1in. thick, fitted with one bolt only

Distance between the centres of girders and stays, 8in.

Boiler pressure, 160lbs. per sq. in.

Applying the Board of Trade formula, we have

$$\frac{660 \times 4{\cdot}625^2\text{in.} \times 1\text{in}}{(16 - 8) \times 8 \times 1{\cdot}33} = 165\text{lbs., the highest working pressure.}$$

The stress on each stay bolt is

$$8 \times 8 \times 160 = 10240\text{lbs.}$$

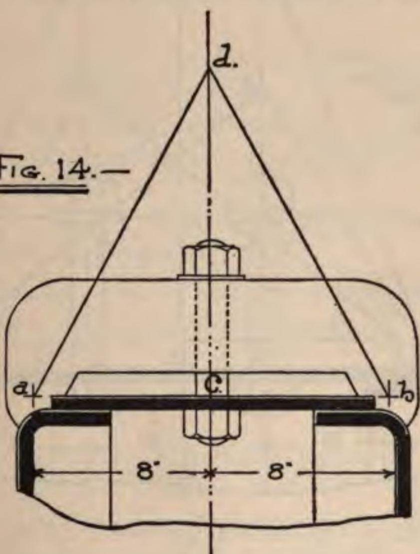
The total bending moment at centre is

$$\frac{10240}{2} \times \frac{16}{2} = 40960\text{lbs.}$$

Then taking the strength of the steel at 61000lbs. sq. in., we have depth² × breadth × radius of gyration × strength = to the total resisting power of the girder, in this case—

$$4{\cdot}625^2 \times 1 \times {\cdot}2886 \times 61000 = 376562 \text{ inch pounds}$$

— FIG. 14.—



The actual stress is 40960lbs.; so if we divide the $\frac{\text{strength}}{\text{strain}} = \frac{376562}{40960} = 9{\cdot}2$ as the factor of safety.

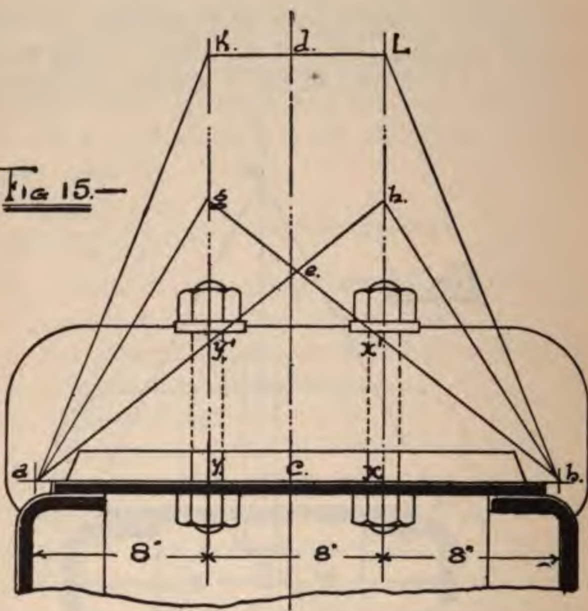
In this case, as in all the others, it is simply a question of leverage. The entire load being in the centre, each point of support takes half the weight, and acts with a length of lever equal to half the width of combustion chamber.

To represent the same result graphically—that is, by lines —if we set off from the point c a length of line $c\ d$ equal to

40960lbs., and join *d b* and *d a,* then the distance *c d* equals total bending moment at centre, and any lines drawn parallel to *c d* cutting the lines *b d* or *d a* will give the proportional bending moment at those points.

Figure 15 shows combustion chamber with two bolts.—

—Fig 15.—



Width of combustion chamber is 24 in.
Girders, $5\frac{3}{4}$ in. × $1\frac{1}{4}$ in., fitted with two bolts.
Distance between centres of girders and stays, 8 in.
Boiler pressure is 160lbs. per square inch.

By Board of Trade formula we have

$$\frac{990 \times 5{\cdot}75^2 \times 1{\cdot}25}{(24 - 8) \times 8 \times 2} = 160\text{lbs., the highest working pressure.}$$

Referring to the figure it will be apparent that each stay bolt has to bear the same stress, because the pressure and leverage are equal, and it is also clear that the centre of girder at *c* will be more severely strained on account of the

increased leverage, and a little patient study will enable the figures to be followed without much trouble.

Comparing our rule with Board of Trade we have—

$8 \times 8 \times 160 = 10240$lbs., the stress on each stay bolt.

$\therefore\ 10240 \times \frac{16}{24} \times 8 = 54613$lbs., actual stress at bolt centres.

And $54613 \times \frac{12}{16} = 40959$lbs., the bending moments caused by each bolt at centre of girder.

Therefore the total stress on centre is equal to the sum of the bending moments, which in this case is $40959 \times 2 = 81918$lbs.

Taking the strength of the steel at 61000lbs. per sq. in., and applying our rule, we have:

Depth ^{2} × breadth × radius of gyration × strength per sq. in. is equal to the resisting power of the girder, which in this case would be:—

$5{\cdot}75^2 \times 1{\cdot}25 \times {\cdot}2886 \times 61000 = 727547$ inch lbs., the total resisting strength of the girder.

The actual stress is 81918 inch lbs., and to resist this we have 727547 inch lbs., therefore dividing the

$\frac{\text{strength}}{\text{strain}} = \frac{727547}{81918} = 8{\cdot}88$, as the factor of safety in this girder.

Representing the stress graphically we have:

Bending moment at x from load at $x = 54613$lbs.

Bending moment at y from load at $y = 54613$lbs.

Draw the lines $x\ h$ and $y\ g$ to any scale, but each in length equal to 54613lbs.; join $h\ b$ and $h\ a$, also $g\ a$ and $g\ b$, then the lines $x\ h$ and $y\ g$ will represent the actual stress at x and y.

Through the point c draw the line $c\ d$ in length equal to the total stress on centre, viz., 81918lbs.

Then $c\ d = 81918$lbs., the stress on centre.

And $x\ h = 54613$lbs., the stress on x.

And $y\ g = 54613$lbs., the stress on y.

From the figure it will be seen that the sum of the bending moments on x and y is equal to the length $y\ y^1 + y\ g$,

which gives $y\ k$. The same applies to x. The length $x\ x^1 + x\ h$ gives $x\ l$, while the bending moment at centre is equal to twice $c\ e$, which gives $c\ d$.

Figure 16 shows combustion chamber with three bolts.

Width of combustion chamber is 32in.
Girders $7\frac{1}{2}$in. × $1\frac{1}{2}$in., fitted with three bolts
Distance between centres of girders and stays is 8in.
Boiler pressure is 160lbs. per square inch.

By Board of Trade formula we have

$$\frac{990 \times 7 \cdot 5^2 \times 1 \cdot 5}{(32 - 8) \times 8 \times 2 \cdot 666} = 163\text{lbs., the highest working pressure.}$$

Applying our rule and comparing it with the above we have

$$8 \times 8 \times 160 = 10240\text{lbs. stress on each bolt.}$$

$$\frac{10240}{2} \times 16 = 81920 = \text{bending moment at } c = 81920\text{lbs.}$$

$10240 \times \frac{24}{32} \times 8 = 61440 =$ the bending moment at y.

$61440 \times \frac{16}{24} =$ its effect at c is equal to .. .. 40960lbs.

$10240 \times \frac{24}{32} \times 8 = 61440 =$ the bending moment at x.

$61440 \times \frac{16}{24} =$ its effect at c is equal to .. .. 40960lbs.

Therefore, the total bending moment is .. ..163840lbs.

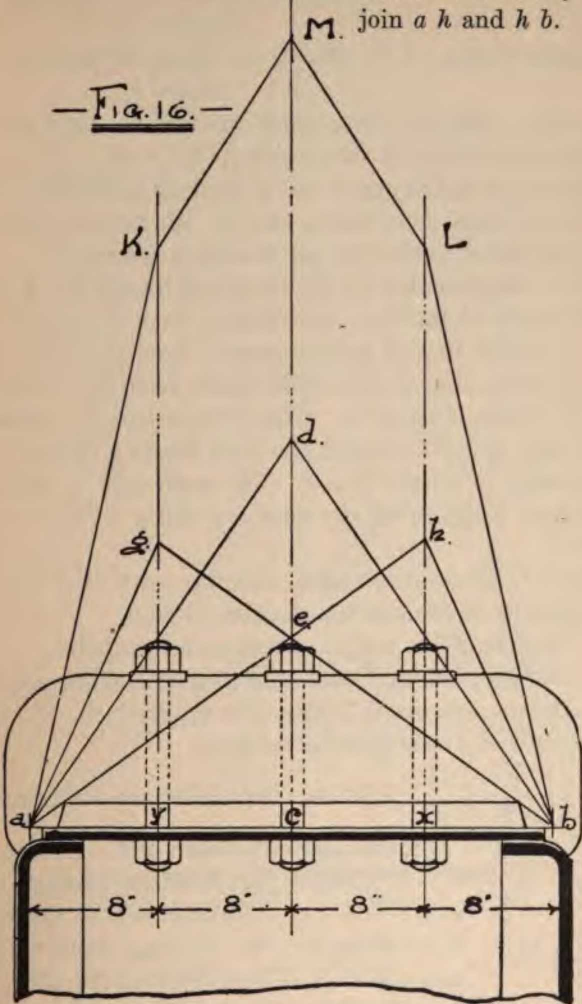
Then $7 \cdot 5^2 \times 1 \cdot 5 \times \cdot 2886 \times 61000 = 1485388$ inch lbs., the resisting power of the girder. The actual load, or total bending moment, is 163840, so dividing

$$\frac{\text{strength}}{\text{strain}} = \frac{1485388}{163840} = 9, \text{ the factor of safety.}$$

To represent this graphically we would have

B.M. at c from load at $c = 81920$, which mark off as an ordinate at c to any scale, join $d\ b$ and $d\ a$.

B.M. at x from load at $x = 61440$, which mark off as an ordinate at x to any scale, join $a\ h$ and $h\ b$.



—Fig. 16.—

B.M. at y from load at $y = 61440$, which mark off as an ordinate at y to any scale, join $b\ g$ and $g\ a$.

Then length of line $c\ d = 81920$lbs., the bending moment at c.
And length of line $y\ g = 61440$lbs., the bending moment at y.

And length of line $x\ h$ = 61440lbs., the bending moment at x.
And length of line $c\ e$ = 40960, the effect of load at y on centre c.
And length of line $c\ e$ = 40960, the effect of load at x on centre c.

Then to $c\ d$ add twice $c\ e$, which will give the total bending moment represented by the length of line $c\ m$.

Referring to the figure, it will be noticed that x and y have each three bending moments, and by adding the length of the three lines together we get the total moments on x and y, which is represented by the length of lines $x\ l$ and $y\ k$.

The length of the first is measured from x to where $g\ b$ cuts the centre line of x; the second, from x to where $d\ b$ cuts the centre line of x; and the third, from x to where $h\ b$ cuts the centre line of x. These three lengths added together give, as before stated, the total bending moment at x, and is equal in length to $x\ l$. The same applies to y, and the stresses being equal the lines $x\ l$ and $y\ k$ are the same length.

Figure 17 shows combustion chamber with four bolts.

Width of combustion chamber is 40in.
Girders, $9\frac{1}{8}$in., × $1\frac{3}{4}$in., fitted with four bolts.
Distance between centres of girders and stays, 8in.
Boiler pressure is 160lbs. per square inch.

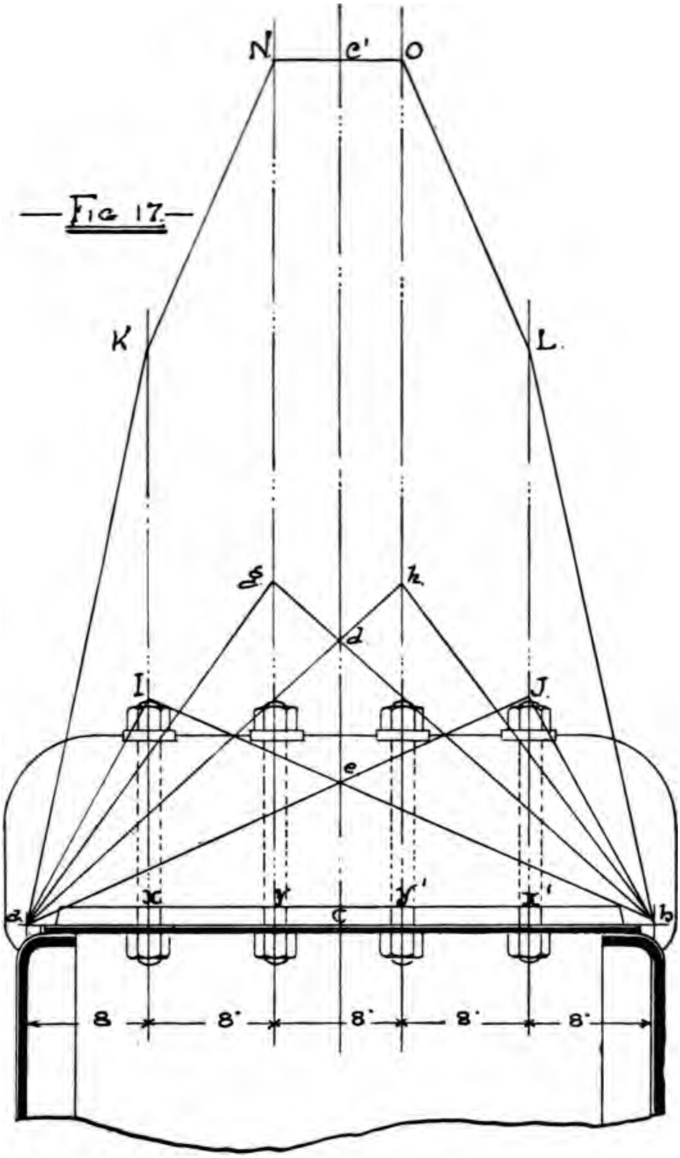
By Board of Trade formula we have

$$\frac{1100 \times 8{\cdot}5^2 \times 1{\cdot}75}{(40 - 8) \times 8 \times 3{\cdot}33} = 163\text{lbs., the highest working pressure.}$$

$8 \times 8 \times 160 = 10240$lbs., stress on each bolt.

$10240 \times \frac{24}{40} \times 16 = 98304$lbs., the bending moment at y;
and $98304 \times \frac{20}{24} = 81920$lbs., the effect at c.
$10240 \times \frac{32}{40} \times 8 = 65536$lbs., the bending moment at x;
and $65536 \times \frac{20}{32} = 40966$lbs., the effect at c.
$10240 \times \frac{24}{40} \times 16 = 98304$lbs., the bending moment at y^1;
and $98304 \times \frac{20}{24} = 81920$lbs., the effect at c.
$10240 \times \frac{32}{40} \times 8 = 65536$lbs., the bending moment at x^1;
and $65536 \times \frac{20}{30} = 40966$, the effect at c.

Then total bending moment at c = 245772 inch pounds.



— Fig 17. —

Applying our rule we have

$8 \cdot 5^2 \times 1 \cdot 75 \times .2886 \times 61000 = 2225829$ inch lbs., the actual resisting power of the girder.

Therefore dividing $\frac{\text{strength}}{\text{strain}} = \frac{2225829}{245772} = 9$ the factor of safety.

To represent this graphically we have the following:—

B.M. at y from load at $y = 98304$, which mark off as an ordinate at y, join $g\ a$ and $g\ b$.

B.M. at x from load at $x = 65536$, which mark off as an ordinate at x, join $I\ a$ and $I\ b$.

B.M. at y^1 from load at $y^1 = 98304$, which mark off as an ordinate at y^1, join $h\ a$ and $h\ b$.

B.M. at x^1 from load at $x^1 = 65536$, which mark off as an ordinate at x^1, join $J\ a$ and $J\ b$.

Then length of the line $y\ g = 98304$, the bending moment at y

" " $x\ I = 65536$ " " x

" " $y^1\ h = 98304$ " " y^1

" " $x^1\ J = 65536$ " " x^1

" " $c\ e = 40966$, the effect of load x on [centre c

" " $c\ e = 40966$, the effect of load x^1 on [centre c

Then to twice $c\ e$ add twice $c\ d$, which will give the total bending moment at c, represented by the length of line $c\ c^1$.

Referring to the figure, it will be noticed that this girder has to bear four variable and distinct bending moments, whose intensity is represented by the distance measured from the top plate of combustion chamber to the points where the four angular lines cut the centre line of each stay bolt (as described in example with three bolts), and that the length of these four lines added together gives the total

bending moment on girder at centre line of each bolt, which in this case is as follows:—

Total bending moment on y is equal to the length of line						y N
"	"	x	"	"	"	x K
"	"	y^1	"	"	"	y^1 O
"	"	x^1	"	"	"	x^1 L
"	"	c	"	"	"	c c^1.

It will be noticed that in the four examples given the *number* of bending moments transmitted to girder increases as the number of bolts increase—that is, for 1, 2, 3, and 4 bolts we have 1, 2, 3, and 4 bending moments.

The graphic method of representing *"stresses"* by lines is coming into general use, and in Foley's excellent book on Boiler Construction the whole of the calculations relating to shells, furnaces, stays, combustion chambers, flat surfaces, girders, tube plates, riveting, etc., are worked out on this system. Line diagrams are constructed for all parts of the boiler, and any required pressure, thickness of shell, diameter, length, or thickness of corrugated or plain surfaces, size or pitch of stays or girders, etc., can be read off without putting a single figure down. This saves an immense amount of work, and to a professional man is invaluable. But such books can never be thoroughly appreciated unless the user knows something of the principles on which such diagrams are constructed. Hence it is that the two methods of calculation are given, as it may prove an incentive to study books where the system is described, explained, and applied.

Respecting the Board of Trade Rule for Girders, it will no doubt be noticed that the factor of safety in all four examples is very high; in fact twice as much as is considered necessary for shells, furnaces, stays, etc.; but there are a number of points that require to be considered and provided for. Taking usual shop practice, we can scarcely depend on the points of support of girders being anything like a perfect or exact fit on the tube plates, as in nearly all cases a certain amount of the strain is developed on and borne

by the top plate itself and, whatever this amount may be, it acts with a small but certain amount of leverage. Then there is the element of uncertainty (as usually fitted) of the stresses being equally distributed, both at the points of support and also in the screwing up of the supporting stays, fitting of washers, corrosive action on the large surface exposed, etc. These and various other things point to the necessity of having an ample margin of strength, which though exceptionally high theoretically, is, in a practical sense, a sensible provision on the right side.

What has been said respecting combustion chamber girders refers to the usual method of staying, and, all things considered, may be put down as the best; but there is a variety in the systems of staying by different makers that might be briefly referred to.

The girders themselves are generally made of two plates. These plates are riveted together with a distance piece to allow the supporting stay bolts, which are firmly screwed to top edge of girder by nuts and substantial washers, to pass freely between them. Sometimes, but not often, the ends of the two plates are welded together and form one point of support at each end instead of two; in other cases, as in locomotives, they are forged solid, with the stay holes drilled and the ends and sides carefully slotted and fitted. Recently a number have been made of cast steel, and no doubt if we could depend on the material, this would be the cheapest and best; but so far, we fight shy of the cast steel for this purpose.

Instead of girders with stays, angle-irons, in single but usually in double shear, are riveted direct, with thimbles between, to the top plate; then between the two angle-irons plates, taking hold of similar double angle-irons at top of shell, are either bolted or riveted in the usual way.

Another plan sometimes used is to flange the top combustion chamber plates, fitting between such flanges vertical plates as webs, while the top of web plates are secured by double angle-irons riveted to shell.

Circular and corrugated tops have in rare cases been fitted, in others the stays are screwed direct into top plate, having a single eye projecting, then a direct stay with double eyes at each end takes hold of the screwed single eye in top plate, the other end to strong tee irons fitted to shell.

Sometimes are fitted what might be termed suspended girders, which do not rest or even touch the top plates at all. The supporting bolts are fitted in exactly the same way as shown, but the whole stress is taken by the suspended stays (generally made square in section) which take hold of the girder by strong bolts in double shear at one end, and at the other end are secured by passing between strong double lugs-angle or tee irons, which are firmly and securely riveted to shell. When suspended stays are used, it is desirable to stay the bottoms of combustion chambers to the bottom of shell, but in double-ended boilers, where the combustion chambers are of exceptional width, the thickness of the tube plates required by the rule would be excessive; and in such cases, it becomes a necessity for these stays to be fitted. This system of staying relieves the tube plates from compressive stress, and therefore allows of much thinner plates being used, which, in the opinion of several leading engineers, would be a radical cure for leaky tubes, especially under forced draught—the theory (since confirmed by experiment) being that it is only reasonable to infer that within certain limits the more nearly the tube and tube plate approach each other in thickness, the more nearly will they expand and contract in harmony, and therefore the chance of leakage will be less; and as we cannot thicken the tubes with advantage, the next best thing to do is to conform to the necessity of making the tube plate as thin as other practical considerations will admit. This was shown in a very practical manner by Mr. A. T. Yarrow in a most valuable paper read before the Institution of Naval Architects, in 1892, in which he proved, by actual experiment, that thick tube plates and thin tubes could not and did not stand the conditions of forced draught

nearly so well as when the thickness was reduced to somewhat near that of the tube, besides having the advantage of a comparatively thin plate to transmit the heat. On the other hand, we have hundreds of boilers with tube plates $\frac{3}{4}$in. and even $\frac{7}{8}$in. thick, which steam continuously for weeks and months with natural and forced draught, carrying 160 and 180lbs. of steam, going all round the world, and doing it over and over again without a hitch. This particular question has two sides to it, and the defect of leaky tubes is principally confined to naval boilers, the mercantile marine being comparatively free from this trouble.

Tube Plates (Steel and Iron).

We now pass on to consider and provide for the compressive stress which tube plates have to bear, and which is developed and transmitted from the combustion chamber girders at their respective points of support.

What we have to be satisfied about is to so construct the tube plate that it will bear the maximum compressive stress and leave a proportional margin of strength. The Board of Trade Rule does all this and leaves an ample factor of safety.

The Rule is as follows *for steel plates*:—

$$\frac{(D - d) \times T \times 28000}{W \times D} = \text{Working pressure.}$$

Where D = least horizontal distances between tube centres in inches.

d = inside diameter of tubes in inches.

T = thickness of tube plates in inches.

W = extreme width of combustion chamber in inches from front of tube plate to back of fire box, or the distance between combustion chamber tube plates when boiler is double-ended and the chamber common to the furnaces at both ends.

Take the following example, which is a fair sample and likely to be found in ordinary practice:—Assume 3½in. tubes, ⅛in. thick, and allowing 1¼in. spaces between them horizontally. By the application of the formula, we would have—

Distance between tube centres	..	4·75in.
Inside diameter of tubes ..	..	3·25in.
Thickness of tube plate ..	..	·75in.
Width of combustion chamber	..	30in.

Find working pressure? Here—

$$\frac{(4{\cdot}75 - 3{\cdot}25) \times {\cdot}75 \times 28000}{30 \times 4{\cdot}75} = 221\text{lbs. per sq. in.}$$

For iron tube plates the Rule is the same, except that the constant is reduced from 28000 to 22000, which, of course, will reduce the working pressure in that ratio; and this is necessary, because steel may be considered for all practical purposes as being of equal strength when exposed to tensile or compressive strains, and is usually taken at 61000lbs. per sq. in. But it is very different in iron, that material being considerably weaker in compression than in tension—hence the reduction in the constant.

Running out the figures for iron, we have—

$$\frac{(4{\cdot}75 - 3{\cdot}25) \times 75 \times 22000}{30 \times 4{\cdot}75} = 173 \text{ lbs. sq. in.}$$

It is very important that all tube plates, where the combustion chamber tops are stayed with girders as before described, should be strong enough and stiff enough for their work; for if not, we may expect bulging, oval holes, and leaky tubes, defects which are not only annoying and expensive, but extremely difficult to rectify.

Tubes.

Although the manufacture of boiler tubes has been, and at present is, a speciality, still in some cases the quality supplied is scarcely good enough for high pressures, especially where forced draught is used. All tubes should be made of

material whose ductility is exceptionally high; they should be of uniform thickness and elasticity, perfectly straight, both ends being thoroughly annealed, as it is only such conditions that will enable each individual tube to take its fair share of the strain. It is of much greater importance than is usually supposed to insist upon the tube ends being perfectly clean; and this is recognised by some makers who now offer to supply tubes with polished ends, so that when expanded the metal in tube and tube plate shall be in clean and close contact. In expanding tubes great care should be taken that when the expanding is finished it does not represent a tapered pin in a parallel hole. In all expanders the rollers should pass through and act upon the entire thickness of tube plate, so that the whole surface is in actual contact—in fact, the usual practice (and especially in the long tubes of locomotives) is to form a small shoulder on the water side of tube plate, which prevents the tube from shifting when a compressive strain comes on, and also ensures all the tubes going and coming together, the expansion and contraction being developed on the large bold radius of the tube plates.

Stay Tubes.

In the British mercantile marine, and also in the Navy, it is compulsory that all tube plates must be fitted with stay tubes, which, if possible, are usually set off in squares, and both stay and surface supported are treated in the same way and the calculations made by the same rules which govern flat surfaces and water space staying.

Although the Board of Trade and Lloyds insist on all tube plates being fitted with stay tubes, yet the opinion is by no means unanimous that they are absolutely necessary—in fact, such men as the late Dr. Kirk, Yarrow, and others hold opposite views. Yarrow's opinion is as follows:—"With regard to stay tubes, for many years we have not used them and have found no injurious result. On the contrary, I believe, so far as the leakage of tubes is concerned, they are

bad. In the first place they are invariably thicker than the surrounding ones, and to have rigid tubes alongside those which are elastic is clearly undesirable. If stay tubes must be adopted, they should be equally elastic with the others and consequently of the same thickness and the same material, only thickened up where they screw into the plates.''

In locomotives carrying high pressures, both in England and America, stay tubes are seldom used, and in American steamers they are not considered necessary.

The experiments which have been carried out to determine the holding power of ordinary tubes are conclusive, *but only under certain conditions.* It takes from 8 to 12 tons to draw a 2in. tube out of an ordinary steel tube plate, the difference between 8 and 12 being probably due to the more or less imperfect fitting in the tube holes; but taking the least—viz., 8 tons—it was shown that for the pressure carried and surface supported, the tube plate had a margin of strength of 20 to 1. So far as locomotives are concerned, this demonstrates that the tube plates are amply stayed by ordinary tubes, and the result of many years' experience in actual working has practically affirmed that stay tubes are not required.

The same conditions, however, do not hold when applied to marine boilers. In locomotives the water is almost always comparatively clean, consequently the internal surfaces (especially tube necks and tube plates) are clean—hence it is the tubes keep tight and give no trouble. In marine boilers it is different; tubes, tube plates, etc., get dirty, and in many cases there is very little time and comparatively few opportunities for cleaning. This results in leaky tubes, which, instead of removing the cause by having them thoroughly cleaned, are expanded and hammered again and again. Many of the ends become hard and brittle and drop off; sometimes a considerable number are flush, and even below the flush, of the face of tube plate, and generally their original holding power is so materially reduced that

were it not for the stay tubes the tube plate would bulge, draw over the tubes, and probably cause a serious accident.

Such things have occurred, and will occur again if stay tubes are not fitted; and, from personal knowledge, in more than one American steamer the above has taken place with fatal results.

Another thing is that if the water by any means got too low the holding power of ordinary tubes would be affected and lessened much sooner than the stay tubes, all of which are stronger, stiffer, and more firmly attached by being screwed into tube plates or secured by double nuts at one or both ends. Again, in all modern marine boilers it almost becomes imperative to use stay tubes to support the front or smokebox tube plate, as in the wings and also in the centre we must have not less than a 10in. space for cleaning, examination, and repairs. That being so, stay tubes of ample section at bottom of thread become a necessity, as ordinary tubes would be weak for the stress to be supported. No one can argue that stay tubes are absolutely necessary in marine, any more than in locomotive boilers. provided the working conditions are the same, which they are not, as has been shown. Condensers will leak and tubes split; air, circulating, and feed pumps have, and will, break down —in fact, a dozen things may take place. Even with the best and most modern machinery it becomes necessary (sometimes) to put salt water into the boilers, and with present pressures a very thin scale about tube necks will very soon produce leaky tubes, which at once affects the holding power of the ordinary tubes.

In the old jet condensing boilers, carrying from 20 to 30lbs., there were no stay tubes and no expanders, but all the tube ends were beaded over. When there was any undue accumulation of salt or scale (which was pretty often) the tubes leaked, and, from the continual hammering trying to make them tight without cleaning them, many of the beadings dropped off altogether, and very often you would find considerable numbers drawn partially through the tube

plate; their holding power was gone, and it became necessary to fit a number of through stay bolts to keep the tube plates in position.

It is, no doubt, much better in modern boilers, but provision must be made for what has happened and may happen again—that is, if ordinary tubes leak, and if such are expanded and hammered several times, then their holding power becomes an uncertain and unknown quantity; therefore, all things considered, the Board of Trade are quite justified in making the fitting of stay tubes compulsory.

Steam Domes.

A few years ago, almost every boiler, both land and marine, was fitted with a steam dome; the generally accepted notion being that it was absolutely necessary to have some such vessel in which the steam could be stored, and that those steam receivers would prevent priming, and would be an advantage to the boiler in every way.

Now we know better, and the fallacy of such constructions (so far as marine practice is concerned) is pretty well exploded, few, if any, modern boilers being fitted with steam domes, chests, or superheaters.

The quantity of steam those structures could hold was in almost every case very fractional indeed; a few strokes of the piston would empty them; in fact, when unclothed and exposed to the air, instead of adding to the boiler's efficiency, they actually lessened it, for the tendency was to act as a partial condenser—the steam taken from them containing more moisture than if the main steam pipe had been connected direct to the steam space of the boiler.

Then as to their preventing priming. This is another absurd idea. Consider how steam is generated, and it will at once be apparent that unless ample provision is made for a sufficiency of area at the water level, the steam cannot and will not leave the water freely or quietly.

Any undue contraction at the surface of the water must naturally result in a tendency to produce priming, and the

fitting of a steam dome as big as a church would not help us much, because it would not remove the cause of the evil; the contracted water level would still be there.

Priming or deflecting plates may have some effect, and lessen the priming, but you are never sure, as in all such the boilers require special attention, and even where the greatest care is exercised they take (and generally without notice) periodical fits, which are always troublesome to the engineer, and sometimes end in material damage to the machinery.

It is now recognised that when designing any boiler, one of the most important considerations is, that the steam space and water surface shall be large enough to enable all the steam generated to leave the water surface quite freely, and that it is much better, more sensible, and cheaper to increase the diameter of shell 9 or 12 inches than to depend upon steam domes for dry steam and as a means of preventing priming.

In land boilers, the practice of fitting domes is not nearly so prevalent as it was; it is gradually but surely dying out; in fact, the only thing that can possibly be said in its favour is that the flat tops are more handy for fitting mountings than the cylindrical shell.

Sometimes the tops are made with spherical or dished ends, and when properly set out (as previously explained), such tops are strong enough—not so the bottoms.

Consider any *portion* of a cylindrical shell; if it is subjected to direct tension, it is necessary that it should be acted upon by a uniform internal pressure and the ends (so to speak) of this cylindrical *portion* be supported by tangential forces equal to the tension on it, but the part of shell under the steam dome does not comply with those conditions.

From the manner in which the forces act, there is nothing to retain the curved form, because there is the same steam pressure on both sides of it. The tendency, therefore, is to straighten itself, it acting merely as a bent stay, and, as it

straightens, the bottom of the dome would naturally try to open and thereby cause a severe strain on the bottom flanges at opposite sides.

To make matters worse, manholes under domes are often unstrengthened and generally, as usually made, represent the weakest part of the whole structure.

When steam domes are fitted, what is required to make this part equally strong is that it should have a pressure equal to the steam pressure uniformly distributed over it. To effect this, it is therefore much better to make all domes with flat tops and stay them with vertical rods passing through shell and strengthening plates.

This will utilise the curved form, take any undue strain off the bottom flanges of dome, and support the flat top at the same time.

It would, however, make a sounder, stronger, and safer boiler if they could be dispensed with altogether.

Respecting the water surface or area of water level in boilers, it should, as before stated, be sufficently large to ensure a free and quiet flow of dry steam without priming; and in reference to this particular part of the subject it is well to remember that the working water level may be reduced in a certain ratio as the working pressure of the steam is increased, because the steam bubbles decrease in size (but not in weight) as the pressure rises.

From this it follows that modern boilers can be worked with considerably less water surface area than in others carrying lower pressures, and the following examples will show the practical application of this.

In marine practice the area of the working water level required for the evaporation of water to steam at any pressure from 50 to 260lbs. (read off the steam gauge) may be found by the following rule, which illustrates the principle, and is approximately correct:—

$$\text{Water surface in square feet} = \frac{2{\cdot}25}{\sqrt{\text{Working pressure}}} \times \text{I.H.P.}$$

Taking three boilers, representing the compound, triple, and quadruple systems, and working at 81, 169, and 256lbs. per square inch respectively. Assume they all develop the same power, viz., 1000 I.H.P.—we want to show what difference there would be in their water level areas.

Applying the rule, we have for the

Compound—

$$\frac{2\cdot25}{\sqrt{81}} = \frac{2\cdot25}{9} \times 1000 = 250 \text{ square feet water level area.}$$

Triple—

$$\frac{2\cdot25}{\sqrt{169}} = \frac{2\cdot25}{13} \times 1000 = 174 \text{ square feet water level area.}$$

Quadruple—

$$\frac{2\cdot25}{\sqrt{256}} = \frac{2\cdot25}{16} \times 1000 = 140 \text{ square feet water level area.}$$

This shows what a material reduction there is as the pressure rises, and that the water surface required decreases as the square root of the pressure increases. This rule may not apply to every boiler, but, unless exceptionally hard worked, it will be found a fair average.

Man-holes and Doors.

Although man-holes and doors are important items in boiler construction, they have in the past, at all events, received but scant attention. Indeed, not many years ago, certain makers, who had a reputation for turning out good work, did in a number of instances cut large holes in the shell circle, and never gave the slightest consideration to the necessity of fitting compensating or strengthening rings to make up for the material lost; and it was a usual thing to find a large majority of such holes cut the wrong way; that is, the longest diameter of the oval was in the direction of the boiler's length instead of being at right angles to it.

This, of course, was not done intentionally, but simply from a want of knowledge of the true principles which

govern construction. Again, it often happened when makers had some idea that something of the kind was required; but it was of a very vague and hazy character, and generally took the shape of riveting a thin, narrow, useless strip of iron round the opening, with holes in it absurdly pitched and so near the edge as to actually weaken it; the section of metal on each side of the rivet in the ring being less than the area of the shell plate punched or drilled out for the rivet. As a rule, man-hole doors were and are now very roughly made and carelessly fitted; very often the camber on doors does not correspond to camber of shell, and when constructed to fit a flat surface are just as likely to have a twist in them as not; in fact, their fitting, fairly or otherwise, in some shops is never even tried, the workman relying in blind confidence on the thickness of the gasket or indiarubber for a tight joint. This has resulted in much trouble, many accidents and not a few explosions, for with unstrengthened holes and badly fitting doors, the tension on the shell plate is certainly not lessened; on the contrary, it is concentrated and increased. This is specially so with leaky joints, which waste and materially weaken the plate.

Where such conditions obtain the plate at the edge of the hole has not only to bear the steam pressure, which holds the door in position, but also the extreme strains, which are invariably put upon it by excessive "screwing up," in the hope (often vain) of making it tight. In such cases it generally results in the edges of the shell plate becoming buckled, and very often fractured, usually at the points where the cross bars bear on it, and which, as before stated, is in many cases the weakest part of the boiler.

Strengthening rings should always be fitted, and their thickness should never be less than that of the plates they cover, while their sectional area should be at least equal to the amount cut out.

They should be well secured to the strake of shell or other plating by a double row of rivets, judiciously pitched, and

well back from the edge of hole; it must, in fact, be a real, and not a sham ring.

Sometimes it is not possible to get a breadth of ring equal to area cut out if made the same thickness. This, however, makes the best job, but where the breadth is limited, owing to rivets, seams, stays, etc., being in the way, the thickness of ring must be increased to make up for the required section. Cast iron domes, or necks, should never be fitted. They are, and must always be, an element of weakness from the fact that the stretching powers of the two metals are very different, cast iron elongating so much less than wrought iron for the same strain, and if the two stretch together the cast iron must of necessity break long before the breaking strain comes on the wrought iron. All man-hole doors should be fitted hot, screwed up, and carefully hammered metal to metal. This should be done before the shell or end plates, as the case may be, are put together.

Another thing productive of much trouble is that in a number of cases the stiffening, or guiding plate, is made much too shallow, and if the gasket or rubber be extra thick there is the difficulty and danger of not getting the door in its proper position. Many fatal accidents have occurred through this, because unless great care is exercised one is apt to get the door high, low, or too much to one side, and when the joint is screwed up the door will catch and jam, thereby making it almost a certainty that when the steam pressure comes on it the joint will be blown out.

In all doors, therefore, the stiffening, or guiding plates, should not only be a decent fit, but they *must* be deep enough to project well through shell or end plates, even with the thickest gasket.

The orthodox size of man-holes is generally 16in. × 12in., and in all cylindrical shells should have their shortest diameter (12in.) placed longitudinally. In the construction of doors the bolts taking the dogs, or cross bars, should

be made of the very best material, and firmly screwed into them with a substantial shoulder, or collar, and if thin with a nut inside, the bolt ends being securely riveted over nut. When well fitted and of ordinary size two bolts are quite sufficient, and for small mud holes (which should also be strengthened) one is enough. The cross bars, especially for bottom man-holes, below furnaces should be forged with a large excess of material, as they waste much faster there owing to cleaning fire, wet ashes, etc.

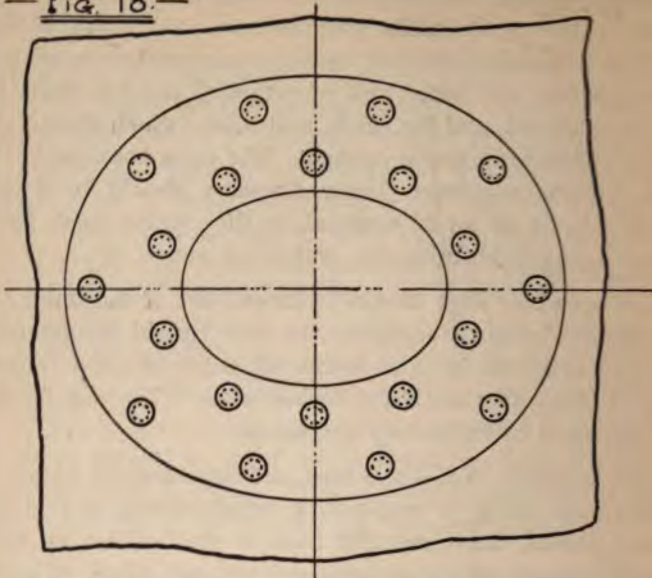
All man-hole dogs should be forged out of the solid, and all holes should be drilled; the feet should be carefully bedded and bear fair and square all over and not on points, which they often do if not looked after. The nuts on the bolts should be extra deep and square.

All holes cut in shells, domes, and superheaters for boiler mountings, such as stop-valves, check-valves, and steam-cocks, should have wrought iron or steel plates or rings riveted round such openings, and all such rings or plates should have a thickness at the thinnest part equal to that of the plate they cover; the top surface, which takes the flange of the cock or valve should have a true turned surface, and as a rule are made considerably wider than the valve flange to which they connect; this ensures ample sectional area and a tight joint. Cast iron should (for reasons previously given) never be used for this purpose.

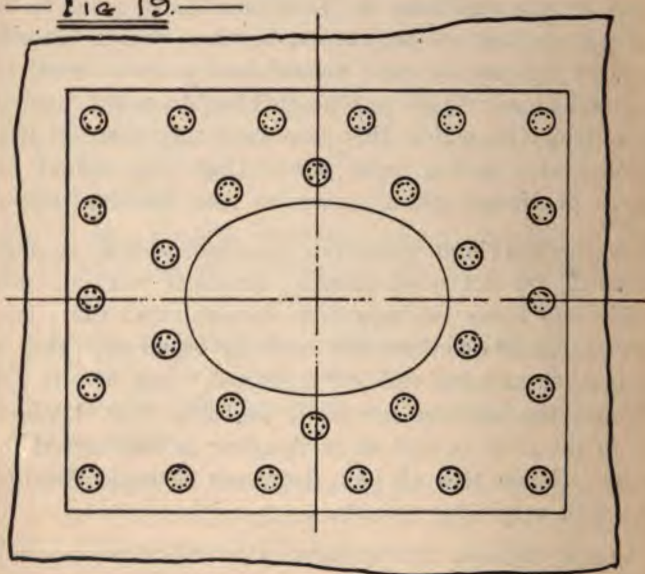
For modern high pressures, man-hole doors, as above described, are not good enough; practical working, much trouble and many mishaps soon demonstrated that; hence all or nearly all doors are now made by machinery; they are stamped, or as some call it "embossed," out of one piece of plate; the surfaces are made perfectly true, while the oval in the ring as well as in the door is also turned in a special lathe, so that all such doors can be made absolutely tight with very little trouble.

Figures 18 and 19 represent the ordinary door with strengthening rings of the required sectional area for the

—Fig. 18.—



—Fig. 19.—



amount cut out, and also show the proper arrangement of rivets. The following will illustrate how the required breadth and section may be found:—

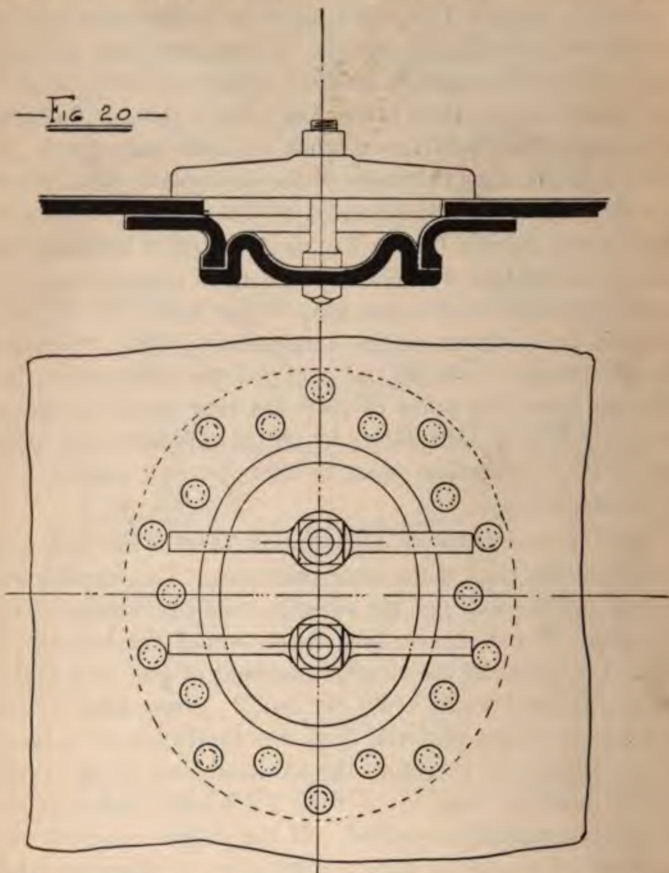
Assume the shell plates to be ½in. thick and the dimensions of manhole 16in. × 12in. We want to compensate for the material drilled or punched out. How much have we lost? Here it is evident the loss must be equal to a strip of plate 12in. long and ½in. thick; therefore 12in. × ·5in. = 6 square inches altogether, and this is what we must make good. If the ring is the same thickness as the shell plate—viz., ½in.—it is clear that its breadth must be 6in., which would give the required section (6in. × 2 × ·5 = 6 square inches); but this section will be weakened by the rivet holes, which will probably reduce its strength, say, 20 per cent. It will still, however, be as strong as the longitudinal joints, which are seldom stronger than 80 per cent. of the solid plate; but although from this point of view the ring would be strong enough, still it is desirable to be on the safe side, and many makers leave a section equal to that cut out, exclusive of the rivets.

Take the same example, and suppose there is not sufficient room to fit the ring of the same thickness. The breadth will be less, but we can get the same section by increasing the thickness. For instance, how much would the breadth be reduced if we make the ring ¾in. instead of ½in., and taking the longitudinal joints at 80 per cent.? Here, 12in. × ·5in. = 6 square inches removed from the shell plate in a longitudinal direction; therefore the sectional area in the ring's breadth must be 3 sq. in. × ·8 = 2·4 square inches of sectional area actually required. If the ring is secured with two rows of rivets, as shown, each $\frac{13}{16}$in. in diameter, then the area of each rivet will be approximately ·5 sq. in.; therefore ·5 × 2 = 1 square inch of rivet section, and 1 + 2·4 = 3·4, which, divided by the thickness, ·75in. = 4½in., the breadth of ring.

A good safe rule, and one easily remembered, is to make the breadth of strengthening rings (when the thickness is

the same as the plates of the cover) equal to *half* the least diameter of man-hole; and where there is plenty of clear space, a good square plate of ample section, as shown in Figure 19, may be fitted with economy and advantage.

—FIG 20—



A fair sample of a modern high pressure door is represented in Figure 20. The strengthening ring is fitted from the inside of boiler, the surfaces are all turned true, and the joint can be made perfectly tight by the same means and in the same way as a cylinder cover. Another thing that recommends this particular door (M'Neil's patent) is—it

can be easily got at and caulked from both sides; there is no straining, and you have a first-class joint on which you can depend with all confidence.

Riveting.

We now pass on to consider the all important question of riveting. There are few mechanics who do not know that a riveted joint is not so strong as the solid plate, although until Fairbairn's time, and even for years afterwards, many were widely astray on this point, and even at present it takes some trouble to convince unreflecting people of the fact.

In punching holes along or near the edge of a plate a little reflection will make it clear that the plate must be weakened in proportion to the amount of material punched out, and that it is impossible to retain the original section of the iron or steel.

Consider a boiler plate unpunched: we have the whole section untouched, unimpaired, and this is usually expressed by saying that the solid plate is always equal to 100 per cent.; but if we drill or punch any holes in it, then this percentage must be reduced. Assume we punch 1in. holes in it, and that the "pitch," or distance, between the centres, is 2in. How much have we lost?

Here it is evident that every alternate inch is a hole 1in. in diameter; therefore the amount of material left between the edge of each hole must be 1in. and no more. So that we have reduced the original strength of the plate one-half; that is, by drilling 1in. holes in any plate *of any thickness* whose centres are 2in. apart we bring the solid plate (equal to 100 per cent.) down to 50 per cent. If we put a double instead of a single row of such holes, *retaining the same pitch,* viz., 2in., would it make the plate section any stronger? No, not a bit stronger, even if we had 10 rows instead of one. Why? Simply because although we increase the number of rows, we punch or drill the same percentage of plate out of each row, consequently the

reduced strength in each row remains the same. The only way to augment the strength is by *increasing the pitch;* thus:—If we make the distance between the hole centres 3in. instead of 2in., it is very evident that the percentage of plate left between the holes must be greater. How much greater?

The holes being 1in. and the pitch 3in., we have punched or drilled out exactly one-third, or 33·3 per cent. of the material, consequently, we must have 66·6 per cent. of plate section left between the holes. If the pitch were 6in., we should lose one-sixth = 16·6 per cent., and 100 — 16·6 = 83·4 per cent. of plate left; and if the pitch were 10in., we should lose $\frac{1}{10}$th, and have 90 per cent. of plate left, so that as above stated, the strength of the plate section depends entirely on the pitch or distance between the hole centres. That it is necessary to direct special attention to this elementary item of instruction is proved by the fact that in this country—in our principal cities, and also in the country districts—it is no exceptional thing to find boilers specified and constructed with two and three rows of rivets in the longitudinal and other seams, whose pitch is only suitable for a single row; that is, two-thirds of the rivets are absolutely useless—one row being as good and equally as strong as two or three. This simply means a waste of good material, more weight, and extrâ expense, without the slightest additional strength.

Having seen how the plate section should be treated, we have also to consider carefully the size and section of rivet. When two plates are riveted together, whatever amount of plate section is left between the holes should be proportional or equal to the sectional area of the rivet —that is, if the plate section be 50, 60, or 70 per cent. of the solid plate, the respective sectional area of the rivets should be the same. This in some cases cannot be arranged to be exactly equal, but there is no difficulty in making the strength of plates and rivets near enough for all practical purposes.

With iron plates and iron rivets, it is usual (although not strictly correct) to assume that the strength of the plate in tension and the strength of the rivet in shear are equal—that is, the tensional force required to break a square inch is equivalent to the force required to shear a square inch. This strength is usually taken as being equal to 47000 lbs. per sq. in.

With steel plates and steel rivets, there is a material difference between the tensional strength of the plate and the shearing strength of the rivet. How much? The plate is generally taken at 28 tons, while the rivet is only equal to 23 tons, per sq. in., which means that if the plate and rivet section be of equal area, the former is 18 per cent. stronger than the latter—($\frac{23}{28} = \cdot 82$, and $82 + 18 = 100$). Consequently with steel plates and steel rivets, to make the joints proportional, the sectional area through the rivets should be 18 per cent. more than the sectional area through the plates—that is, if the plate section is 82 per cent., the rivet section should be 100 per cent., because $100 \times \frac{23}{28} = 82$.

In exceptional cases rivets are subjected to a tensional strain only, as in steam space staying, where the stay ends are secured between double angle-irons, etc., but nearly all rivets have to bear a shearing strain generally. In all lap joints the rivets are in single shear—that is, the shearing strain can only act on the sectional area of the rivet, and no more. It can only shear in one place, but in all joints fitted with double butt straps (one on each side) the rivets are in double shear, and if properly fitted must shear in two places, the area to be shorn being doubled. Theoretically, the strength should be doubled also, but practical experiment has demonstrated that we only get 1·75 instead of 2, as the average value of double shear, and this decrease of strength is assumed to be caused (as previously explained) by the rivet being subjected to a severe shearing and tensile strain at the same time, excessive hammering, contraction in cooling, etc.

In engine and bridge work, when double eyes and links

are truly bored and the pins carefully turned, fitted and *only* exposed to shearing strain, we get approximately the theoretical value of the double shear; but the work must be very accurate, of the highest class, and the strain equally distributed.

There is considerable variety in the arrangement of boiler joints, there being no less than from 20 to 30 different and distinct methods of riveting, but in ordinary practice the number may be reduced to 16, which will embrace all that is necessary for the present purpose. The various rules which apply to and govern boiler riveting may, and no doubt will, appear at first sight somewhat complicated and confusing, but the working out of practical examples for each individual joint, will, it is hoped, be sufficient to enable any mechanic of average capacity to not only understand, but to apply the rules to his daily work.

Cylindrical Boiler Shells—Joints with Drilled Holes.

Formulæ for ordinary zig-zag riveted and chain riveted joints, and also for joints of these descriptions when every alternate rivet in the outer, or in the outer and inner rows has been omitted (Board of Trade Notation):—

Let E = distance from edge of plate to centre of rivet in inches.

„ V = distance between rows of rivets in inches.

„ V^1 = distance between inner and middle row of rivets for joints J and K (Figs. 35 and 36).

„ B = boiler pressure in lbs. per square inch.

„ C = 1 for lap or single butt strap joints.

„ C = 1·75 for double butt strap joints.

„ d = diameter of rivets in inches.

„ D = diameter of boiler in inches (inside).

„ F = factor of safety for shell plates.

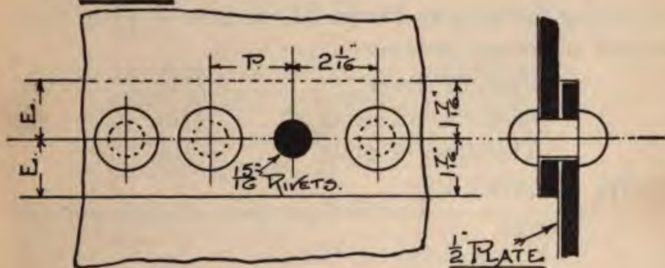
„ n = number of rivets in one pitch.

Let p_D = diagonal pitch in inches.
,, P_D = diagonal pitch in inches between inner and middle rows of rivets for joint J.
,, p = greatest pitch of rivets in inches.
,, r = percentage of plate left between holes in greatest pitch.
,, R = percentage of rivet section.
,, R_1 = percentage of combined plate and rivet section.
,, S = tensile strength of material in lbs. per sq. in. of section.
,, S_1 = tensile strength of plates in tons.
,, T = thickness of plate in inches.
,, T_1 = thickness of each butt strap in inches.
,, % = least value of r, R, R_1, as the case may be ÷ 100.

Examples.

All the drawings are for steel plates and steel rivets.

—FIG. 21.—



Single Riveted Lap Joint.

Referring to Figure 21, we have a single-riveted lap-joint, which represents the most simple method of riveting. To calculate its strength, we must measure three things, viz.:—Pitch of rivets, diameter of rivets, and thickness of plates.

The *"Plate Section Formula"* for iron plates and iron rivets, or steel plates and steel rivets, is the same, viz.:—

$$\frac{(p - d) \times 100}{p} = r \text{ the per cent. of plate section,}$$

and by referring to the notation will be found to mean that the

$$\frac{(\text{Pitch} - \text{diameter of rivet}) \times 100}{\text{Pitch}} = \text{per cent. of plate section.}$$

For the rivet section in iron plates and iron rivets, we have

$$\frac{d^2 \times \cdot 7854 \times n \times 100}{p \times \mathrm{T}} = \mathrm{R} \text{ the per cent. of rivet section,}$$

which, according to the notation, means that the

$$\frac{\text{Area of rivet} \times \text{No. of rows} \times 100}{\text{Pitch} \times \text{thickness}} = \text{per cent. of rivet section.}$$

When plates and rivets are of steel, we have to allow for the difference between the tensile strength of the plate and the shearing strength of the rivet, which is as 28 tons to 23 tons; then the formula becomes (*for single shear*)

$$\frac{d^2 \times \cdot 7854 \times n \times 100 \times 23}{p \times \mathrm{T} \times 28} = \mathrm{R} \text{ the per cent. of rivet section.}$$

Applying the rules to Figure 21, we have in plain figures (for steel plates and steel rivets) :—

Pitch of rivets ..	..	$2\frac{1}{16}$ in.
Diameter of rivets	..	$\frac{15}{16}$in.
Thickness of plate	..	$\frac{1}{2}$in.

$$\frac{2 \cdot 0625 - \cdot 9375 \times 100}{2 \cdot 0625} = 54 \cdot 5 \text{ per cent. of plate section.}$$

and

$$\frac{\overset{\text{area.}}{\cdot 69} \times \overset{\text{row.}}{1} \times 100 \times 23}{2 \cdot 0625 \times \cdot 5 \times 28} = 55 \text{ per cent. of rivet section.}$$

The distance between edge of plate and centre of rivet is $1\frac{7}{16}$ in., which gives us an amount of material from edge of rivet-hole to edge of plate of about $\frac{1}{16}$in. in excess of the rivet's diameter, and in no case should this dimension be less than the rivet's diameter.

If the plates and rivets had been of iron and *we retain the same pitch*, it is evident we can get a slightly increased

percentage of strength with a smaller rivet, and the joint would come out as follows:—

Pitch of rivets ..	..	$2\frac{1}{16}$ in.
Diameter of rivets	..	$\frac{7}{8}$in.
Thickness of plates	..	$\frac{1}{2}$in.

$$\frac{2{\cdot}0625 - {\cdot}875 \times 100}{2{\cdot}0625} = 57{\cdot}5 \text{ per cent. of plate section,}$$

and

$$\frac{\overset{\text{area.}}{{\cdot}6} \times \overset{\text{row.}}{1} \times 100}{2{\cdot}0625 \times {\cdot}5} = 58 \text{ per cent. of rivet section.}$$

Assume we have a steel boiler 72in. inside diameter and $\frac{1}{2}$in. thick, and that the longitudinal joints are riveted, as in Figure 21. At what pressure would we work it?

Taking the steel at 28 tons (62720lbs. per square inch), the strength of joint at 54 per cent., and a factor of safety of $5\frac{1}{2}$, we would have

$$\frac{62720 \times {\cdot}54 \times ({\cdot}5 \times 2)}{72 \times 5{\cdot}5} = 85\text{lbs. per sq. in., W.P.}$$

Had this boiler been of iron we would have—

Strength of iron, 47000lbs. per sq. in.
Strength of joint, 57 per cent. of solid plate.
Factor of safety, $5\frac{1}{2}$.

And —

$$\frac{47000 \times {\cdot}57 \times ({\cdot}5 \times 2)}{72 \times 5{\cdot}5} = 67\text{lbs. per sq. in., W.P.}$$

Expressing the above in accordance with the notation formula it would be

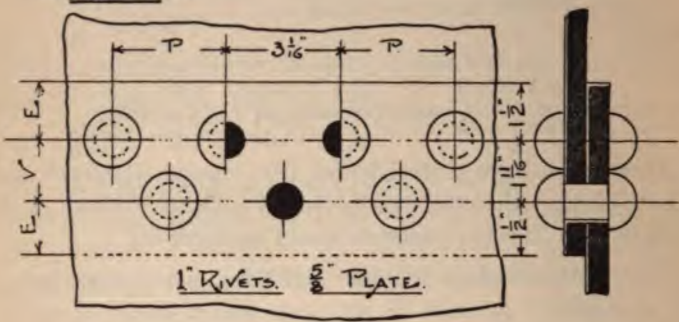
$$\frac{S \times \% \times (2 \times T)}{D \times F} = B, \text{ the working pressure.}$$

Referring again to the figure 21, it will be observed that in all single riveted lap joints there is only one rivet in the pitch, as shown in the shaded rivet; also, that its sectional area must be made equal, or, at all events, approximately equal, to the percentage of plate section left between the holes.

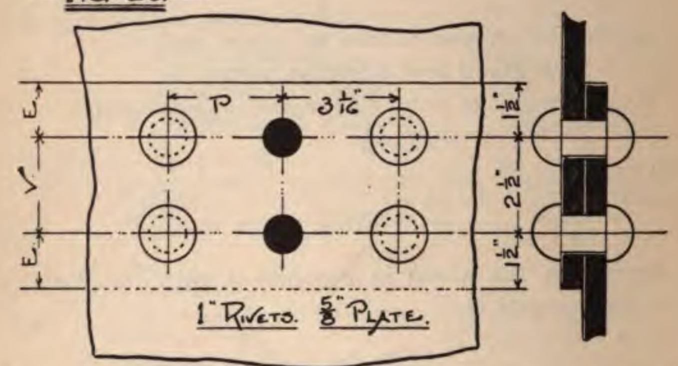
Double Riveted Lap Joint, Zig-zag and Chain.

Figures 22 and 23 represent two double riveted lap joints, in which the pitch, diameter of rivets, and thickness of plates are the same in both, consequently, the strength of

— FIG 22. —



— FIG 23. —



the two joints is exactly equal, but the style or design in the arrangement of the riveting is different and distinct.

This is usually expressed by saying that one is zig-zag and the other chain riveted.

Although the respective strengths are equal, it is the usual practice in boiler work to prefer zig-zag to chain for two

reasons—1st, because for tightness of joint, the rivet section is better distributed, and 2nd, because we require less material, as the breadth of lap is considerably reduced.

In zig-zag riveting the amount of metal measured diagonally between the edges of rivet holes should be from 27 to 30 per cent. greater than *half* the metal between two holes in the horizontal or main pitch, otherwise the plate will be weakest through the diagonal pitching. The Board of Trade Rule for Diagonal Pitch *in this class of joint* is

$$\frac{6 \times p + 4 \times d}{10} = p\text{D},$$

which means that 6 times the main pitch plus 4 times the diameter of rivet ÷ 10 gives the diagonal pitch. This, however, is a minimum rule which scarcely leaves sufficient metal, and might be improved by making diagonal pitch equal to main pitch × ·65 + ·35 of diameter of rivet.

In chain riveting, the distance between the rows of rivets *in this class of joints* is found by the following:— $\frac{4 \times d + 1}{2}$ = the distance between the rows, measured between centres of rivets, and in no case is V to be less than twice the diameter of rivet.

Running out the strength of Figures 22 and 23, we have in plain figures, for steel plates and rivets—

Pitch of rivets ..	..	$3\frac{1}{16}$ in.
Diameter of rivets	..	1in.
Thickness of plate	..	$\frac{5}{8}$in.

Then $\frac{3{\cdot}0625 - 1 \times 100}{3{\cdot}0625} = 67\ \%$ of plate section,

And $\frac{\overset{\text{area.}}{\cdot 7854} \times \overset{\text{rows.}}{2} \times 100 \times 23}{3{\cdot}0625 \times \cdot 625 \times 28} = 67{\cdot}4\ \%$ of rivet section.

If the plates and rivets had been of iron *retaining same pitch,* we can, as in the single lap-joint, get a slightly in-

greater percentage of strength with a smaller rivet, and the joint would come out as follows:—

Pitch of rivets . . .	$3\frac{1}{16}$ in.
Diameter of rivets .	$\frac{15}{16}$ in.
Thickness of plate . .	$\frac{5}{8}$ in.

$$\frac{(3{\cdot}0625 - {\cdot}9375) \times 100}{3{\cdot}0625} = 69\ \%\ \text{of plate section.}$$

$$\frac{{\cdot}69 \times 2 \times 100}{3{\cdot}0625 \times {\cdot}625} = 72\ \%\ \text{of rivet section.}$$

Assuming we had a steel boiler 72in. inside diameter and $\frac{5}{8}$in. thick, and that it was riveted as in Figure 22, at what pressure should it be worked?

Strength of the steel	. . 62720lbs. per. sq. in.
Strength of joint (least) . .	67 % of solid plate
Factor of safety	$5\frac{1}{2}$.

Then:—

$$\frac{62720 \times {\cdot}67 \times {\cdot}625 \times 2}{72 \times 5{\cdot}5} = 132\text{lbs. working pressure.}$$

If the boiler had been of iron, we would have

Strength of the iron	. . 47000
Strength of joint (least)	69 % of solid plate
Factor of safety	$5\frac{1}{2}$.

$$\frac{47000 \times {\cdot}69 \times {\cdot}625 \times 2}{72 \times 5{\cdot}5} = 102\text{lbs. working pressure.}$$

Expressing the above by formula, we have

$$\frac{S \times \% \times 2 \times T}{F \times D} = \text{B, the boiler pressure.}$$

Referring again to the Figures 22 and 23, it should be noted in the zig-zag joint there are two rivets in each pitch —one whole rivet and two halves—and that the shaded parts show very clearly what each rivet has to do. As in the single lap-joint, the combined sectional area of the rivets in each pitch is approximately equal to the percentage of plate section left between the holes. In the chain riveted joint (Fig. 23), it is precisely the same, only the riveting

is differently arranged. There are two *whole* rivets in each pitch, and the distance between the rows of rivets is

$$\frac{4 \times d + 1}{2} = V$$

in this case:—

$$\frac{4 \times 1 + 1}{2} = 2 \cdot 5\text{in. between the rows.}$$

In all the drawings given in connection with riveted joints, the *letters* of the formula are placed at one end and the plain figures at the other, so that any one can see at a glance what the actual dimensions are. This will assist the student in becoming familiar with the notation, and fix in his mind what the respective letters represent.

Treble Riveted Lap Joint, Zig-zag and Chain.

Figures 24 and 25 represent the ordinary treble riveted lap joint fôr steel plates and steel rivets.

Both joints are exactly the same strength, the only difference being in the arrangement of the riveting.

Double and treble riveted lap joints are very similar, but by having three rows of rivets instead of two, we are enabled to widen the pitch and increase the rivet section in proportion, so that the percentage through plates and rivets gives us a stronger joint.

Running out the figures for the zig-zag riveting, as given in figures 24 and 25 we have:—

Pitch of rivets ..	..	4in.
Diameter of rivets	..	$1\frac{1}{16}$ in.
Thickness of plate	..	$\frac{3}{4}$in.

$$\text{Then } \frac{4 - 1 \cdot 0625 \times 100}{4} = 73 \cdot 4 \text{ per cent. plate section,}$$

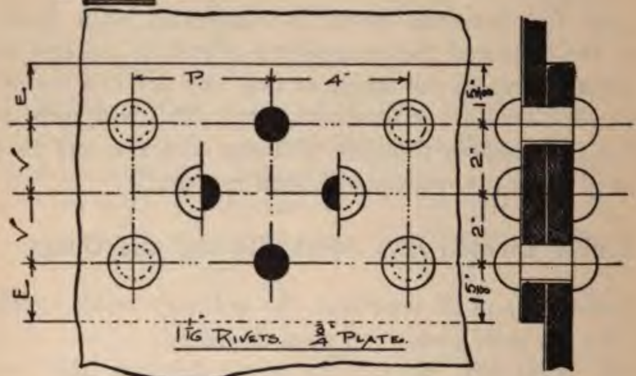
$$\text{And } \frac{\overset{\text{area.}}{\cdot 886} \times \overset{\text{rows.}}{3} \times 100 \times 23}{4 \times \cdot 75 \times 28} = 72 \cdot 7 \text{ per cent. rivet section.}$$

If the plates and rivets had been of iron, we could (as shown in previous examples) get a slight increase in the

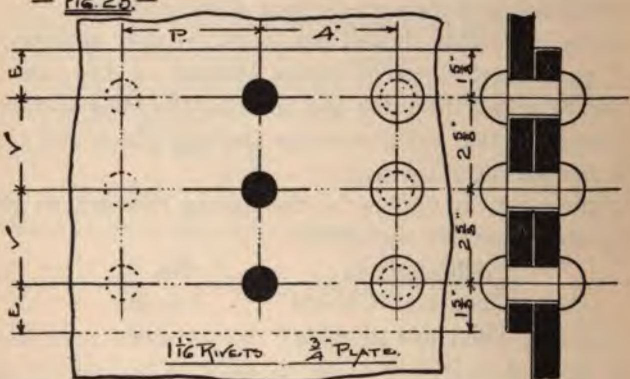
percentage with a smaller rivet, provided we retain the same pitch, and the joint would come out as follows:—

Pitch of rivets ..	..	4in.
Diameter of rivets	..	1in.
Thickness of plates	..	$\frac{3}{4}$in.

—Fig 24.—



— Fig. 25.—



Then $\dfrac{4 - 1 \times 100}{4} = 75$ per cent. of plate section,

And $\dfrac{\overset{\text{area.}}{\cdot 7854} \times \overset{\text{rows.}}{3} \times 100}{4 \times \cdot 75} = 78$ per cent. of rivet section.

In the chain riveted joint (Fig. 25), we must see that V (the distance between the rows) is correct, which in this case is

$$\frac{4 \times 1 + 1}{2} = 2{\cdot}5\text{in. the same as in the double lap.}$$

The working pressure of a steel boiler shell riveted as above, assuming it to be 10 feet inside diameter, ¾in. thick, and riveted as in Fig. 24, would be :—

Strength of steel ..	62720
Strength of joint (least)	73 per cent. of solid plate
Factor of safety ..	5½

$$\text{Then } \frac{62720 \times {\cdot}73 \times ({\cdot}75 \times 2)}{120 \times 5{\cdot}5} = 104\text{lbs. working pressure.}$$

Had this boiler been of iron, then :—

Strength of iron ..	47000
Strength of joint (least)	75 per cent. of solid plate
Factor of safety ..	5½

Then

$$\frac{47000 \times {\cdot}75 \times ({\cdot}75 \times 2)}{120 \times 5{\cdot}5} = 80\text{lbs., working pressure.}$$

In the treble riveted lap-joints it will be observed that they are exactly the same strength, and that in the zig-zag the rivet section in the main pitch is equal to two whole and two half rivets, while in the chain this section is equal to three whole rivets, as shown by the shaded parts on drawing.

Quadruple Riveted Lap Joint.

Quadruple riveted lap joints are used in girder, bridge and ship work, but seldom or never in boiler construction. They are set off and proportioned in the same way as in double and treble lap, the only distinction being that having four rows of rivets we can get a wider main pitch, which enables us to increase the strength percentage of plate section, and by having a proportional rivet section we get a stronger joint.

EXAMPLE.—STEEL.

Pitch of rivets ..	..	$4\frac{1}{16}$ in.
Diameter of rivets	..	$\frac{7}{8}$ in.
Thickness of plates ..		$\frac{5}{8}$ in.

$$\frac{4 \cdot 0625 - \cdot 875 \times 100}{4 \cdot 0625} = 78 \cdot 4 \text{ per cent. plate section.}$$

area. rows.

$$\frac{\cdot 6 \times 4 \times 100 \times 23}{4 \cdot 0625 \times \cdot 625 \times 28} = 77 \cdot 7 \text{ per cent. of rivet section.}$$

Had plates and rivets been of iron the strength percentage would be fractionally more, with a less diameter of rivet and a slight increase in main pitch.

Pitch of rivets ..	..	$4\frac{1}{8}$ in.
Diameter of rivets	..	$\frac{13}{16}$ in.
Thickness of plates	..	$\frac{5}{8}$ in.

$$\text{Here } \frac{4 \cdot 125 - \cdot 8125 \times 100}{4 \cdot 125} = 80 \cdot 3 \text{ per cent. of plate section}$$

$$\text{And } \frac{\cdot 518 \times 4 \times 100}{4 \cdot 125 \times \cdot 625} = 80 \cdot 6 \text{ per cent. of rivet section.}$$

This practically finishes the ordinary lap-joints, in which it will be seen that all the rivets are in single shear. Also that single and double shear is represented in the notation formula by the letter C, but as C in lap jointing is equal to unity, or 1, it is not necessary to introduce it into calculations for this class of joint.

We have now to consider "Butt Strap Joints," in which the rivets may be in single, but are almost always in double shear. All the drawings are for steel plates and steel rivets.

Single Riveted Double Butt Strap Joint.

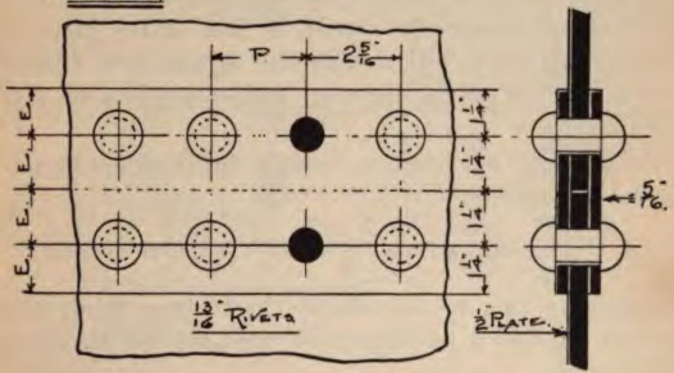
Figure 26 shows a double butt strap joint, single riveted—that is, there is one row of rivets on each side of butt, and its strength is ascertained in the same way as in lap-joints, only that we have to take into consideration the fact that

all the rivets are in double shear, which, of course, materially affects the plate and rivet section. Referring to the Figure, we have

Pitch of rivets ..	..	$2\frac{5}{16}$ in.
Diameter of rivets	..	$\frac{13}{16}$ in.
Thickness of plate	..	$\frac{1}{2}$ in.

Find per cent. strength through plates and rivets which are all steel.

—Fig. 26.—



Here $\dfrac{2{\cdot}3125 - {\cdot}8125 \times 100}{2{\cdot}3125} = 64$ per cent. of solid plate.

and $\dfrac{\overset{\text{area.}}{{\cdot}518} \times \overset{\text{row.}}{1} \times 100 \times 23 \times 1{\cdot}75}{2{\cdot}3125 \times {\cdot}5 \times 28} = 64$ per cent. of rivet section.

The same joint in iron would be

Pitch of rivets ..	..	$2\frac{5}{16}$ in.
Diameter of rivets	..	$\frac{3}{4}$ in.
Thickness of plate	..	$\frac{1}{2}$ in.

Then $\dfrac{2{\cdot}3125 - {\cdot}75 \times 100}{2{\cdot}3125 \times {\cdot}5} = 67{\cdot}6$ per cent. of solid plate.

and $\dfrac{\overset{\text{area.}}{{\cdot}44} \times \overset{\text{row.}}{1} \times 100 \times 1{\cdot}75}{2{\cdot}3125} = 66{\cdot}6$ per cent. rivet section.

It must be remembered that in all the foregoing the value of single shear was unity—that is, T was equal to 1—but when rivets are in double shear then T is equal to 1.75; hence its introduction in the above examples. The number of rivets in the pitch in this example is the whole number in each row as shown.

Butt Straps.

For steel plates and steel rivets, and also for iron plates and iron rivets, the rule is that for this class of joint the thickness of butt straps should be [illegible] of the plate they cover, when fitted with double butt straps; and when the butt straps are single, the strap's thickness is [illegible] of the plate it covers.

According to our notation formula, the thickness would be expressed thus:—

$$\text{Double butt straps } \frac{5\ T}{8} = T_1, \text{ the thickness.}$$

Applying this to our double butt strap joint, single riveted, as shown in Figure 26, we have for the strap's thickness—

$$\frac{5 \times .5}{8} = .3125\text{in. or } \tfrac{5}{16}\text{in. thick.}$$

Had this joint been fitted with a single, instead of a double, butt strap, then by formula—

$$\frac{9\ T}{8} = T_1, \text{ the thickness.}$$

and in this case would be $\frac{9 \times .5}{8} = .5625\text{in. or } \frac{9}{16}$ in. thick.

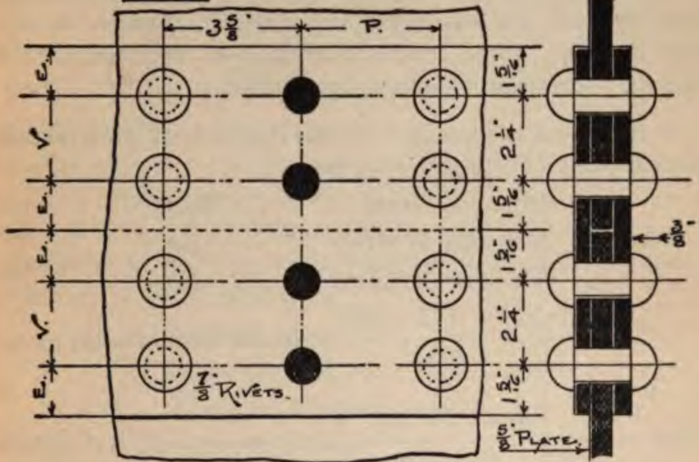
Double Riveted Double Butt Strap Joint, Zig-zag and Chain.

Referring to Figures 27 and 28, we have two double riveted *double* butt strap joints of exactly the same strength, only

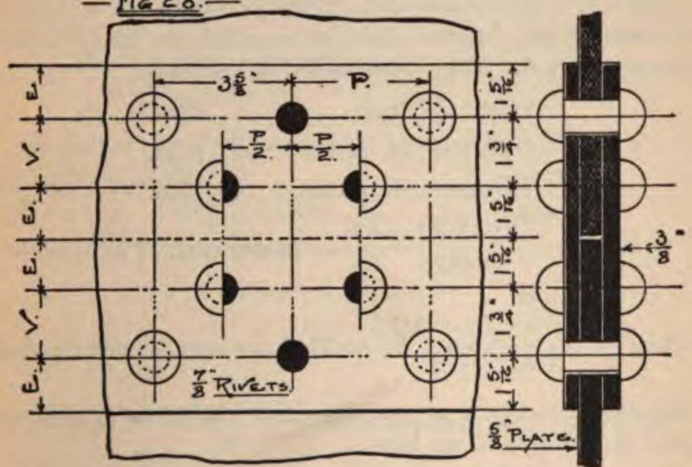
(as in the other examples) one is zig-zag and the other chain riveted.

To a practical eye the zig-zag riveting must commend itself

—FIG 27.—



—FIG 28.—



as being much more likely to make a good and tight joint than the chain riveting, and we also get the same strength with less material. By double strapping boiler joints we

are enabled (through the rivets being in double shear) to increase the rivet section considerably in excess of that obtainable in double riveted lap, and we can do this with a smaller rivet, which also means a wider main pitch, and consequently a proportional increase in the percentage of plate section, besides having the shell, or furnace, as the case may be, more truly cylindrical—a very important item in these days of extreme pressures.

Running out the strength for the double butt strap double riveted joint (Figure 27), we have

Pitch of rivets ..	..	$3\frac{5}{8}$in.
Diameter of rivets	..	$\frac{7}{8}$in.
Thickness of plate	..	$\frac{5}{8}$in.

$$\text{Here } \frac{3{\cdot}625 - {\cdot}875 \times 100}{3{\cdot}625} = 76 \text{ per cent. of solid plate.}$$

$$\text{And } \frac{\overset{\text{area.}}{{\cdot}6} \times \overset{\text{rows.}}{2} \times 100 \times 1{\cdot}75 \times 23}{3{\cdot}625 \times {\cdot}625 \times 28} = 76{\cdot}3 \text{ per cent. of rivet section}$$

With iron plates and iron rivets we would get a slightly stronger joint, because the rivets could be smaller, and consequently the main pitch would be increased.

Pitch of rivets ..	..	$3\frac{3}{4}$in.
Diameter of rivets	..	$\frac{13}{16}$ in.
Thickness of plates	..	$\frac{5}{8}$in.

$$\text{Here } \frac{3{\cdot}75 - {\cdot}8125 \times 100}{3{\cdot}6875} = 78 \text{ per cent. of solid plate}$$

$$\text{And } \frac{\overset{\text{area.}}{{\cdot}518} \times \overset{\text{rows.}}{2} \times 100 \times 1{\cdot}75}{3{\cdot}75 \times {\cdot}625} = 77{\cdot}4 \text{ per cent. of rivet section}$$

Regarding the butt straps, the same thickness applies for steel and iron, and we have:—

$$\frac{5 \times {\cdot}625\text{in.}}{8} = {\cdot}39\text{in. thick or } \tfrac{3}{8}\text{in. full.}$$

Treble Riveted Double Butt Strap Joint, Zig-zag and Chain.

Figures 29 and 30 show the ordinary treble riveted double butt strap joints—one zig-zag and the other chain riveted. They represent the strongest joint used in boiler construction, when the rivets are *evenly* pitched, the strength through plates and rivets being approximately 80 per cent. of solid plate.

The calculation is similar to the double riveted butt strap, only we have three instead of two rows of rivets on each side of the butt. As in the other examples, both joints are precisely the same strength, and taking the zig-zag riveting, we have in plain figures:—

Pitch of rivets .. .. $4\frac{7}{8}$in.
Diameter of rivets .. $\frac{15}{16}$ in.
Thickness of plates .. $\frac{3}{4}$in.

$$\text{Then } \frac{4{\cdot}875 - {\cdot}9375 \times 100}{4{\cdot}875} = 80 \text{ per cent. of solid plate}$$

$$\text{And } \frac{\overset{\text{area.}}{{\cdot}69} \times \overset{\text{rows.}}{3} \times 100 \times 1{\cdot}75 \times 23}{4{\cdot}875 \times {\cdot}75 \times 28} = 81 \text{ per cent. of rivet section}$$

If plates and rivets had been of iron the joint might be arranged as follows:—

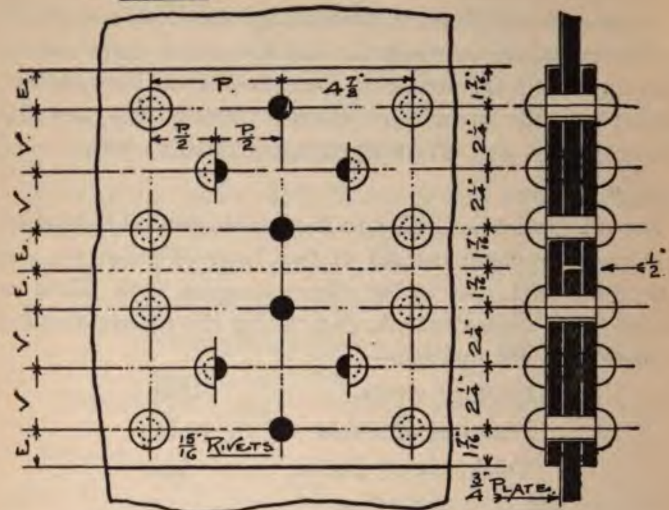
Pitch of rivets .. .. 5in.
Diameter of rivets .. $\frac{7}{8}$in.
Thickness of plate .. $\frac{3}{4}$in.

$$\text{Then } \frac{5 - {\cdot}875 \times 100}{5} = 82{\cdot}5 \text{ per cent. of solid plate}$$

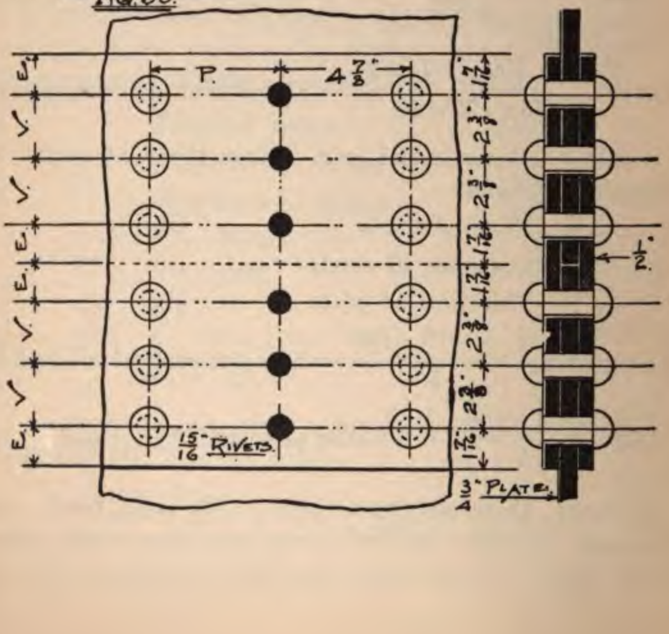
$$\text{And } \frac{{\cdot}6 \times 3 \times 100 \times 1{\cdot}75}{5 \times {\cdot}75} = 84 \text{ per cent. of rivet section.}$$

As shown, there are three rivets in the main pitch, two whole and two halves in the zig-zag, and three whole rivets in the chain. In this joint also the butt straps are the

—Fig 29.—



—Fig. 30.—



same thickness for steel and iron. Therefore in both cases we would have

$$\frac{5 \times \cdot 75}{8} = \cdot 47\text{in. thick, or, say, } \tfrac{1}{2}\text{in.}$$

This practically finishes what is understood by ordinary riveting, as usually met with in boiler construction, and the value of the various joints may be approximately set down as follows (for steel plates and steel rivets) :—

Single riveted lap	54 per cent. of solid plate.
Double riveted lap	68 ,, ,, ,, ,, ,,
Treble riveted lap	73 ,, ,, ,, ,, ,,
Quadruple riveted lap ..	78 ,, ,, ,, ,, ,,
Single riveted butt strap ..	62 ,, ,, ,, ,, ,,
Double riveted butt strap ..	75 ,, ,, ,, ,, ,,
Treble riveted butt strap ..	80 ,, ,, ,, ,, ,,
Quadruple riveted butt strap	83 ,, ,, ,, ,, ,,

When the plates and rivets are of iron the joints are from two to three per cent. stronger, for reasons previously explained, while the difference in the tensile strength of the steel or iron may be approximately taken and expressed as 62720 is to 47000lbs. per sq. in.

Riveted Joints.—Chain and Zig-zag.

Examples of joints in which every alternate rivet is omitted in the outer row, or in the outer and inner rows. All the drawings are for steel plates and steel rivets.

In all such joints the rivets are unevenly pitched. They are generally known as "high per centage" or modern joints, while their design, arrangement, and strength differs materially from the ordinary riveting already described.

Double Riveted Double Butt Strap Joint.

Take Figures 31 and 32, representing double riveted, double butt strap joints, in which the "pitch" of the outer row is exactly double the pitch in the inner row. We want to ascertain their strength. *Assuming all the other dimen-*

sions to be correctly proportioned, the calculation is carried out in precisely the same manner as in the ordinary double riveted, double butt strap joint, and for zig-zag riveting is as follows:—

FIG. 31.

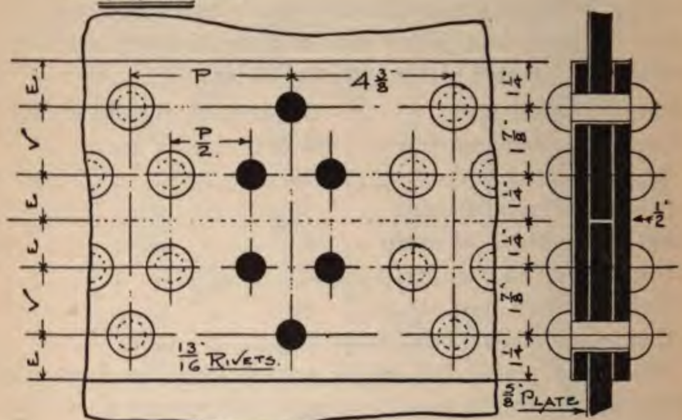
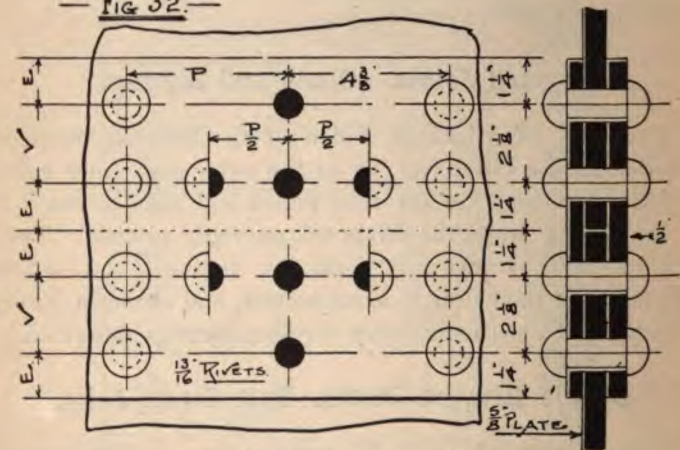


FIG. 32.



Pitch of rivets .. $4\frac{3}{8}$in.
Diameter of rivets $\frac{13}{16}$ in.
Thickness of plates $\frac{5}{8}$in.

Here $\dfrac{4 \cdot 375 - \cdot 8125 \times 100}{4 \cdot 375} = 81 \cdot 4$ per cent. of solid plate.

And

$$\frac{\overset{\text{area.}}{\cdot 518} \times \overset{\text{rivets.}}{3} \times 100 \times 1 \cdot 75 \times 23}{4.375 \times \cdot 625 \times 28} = 81.77 \text{ per cent. of rivet section.}$$

According to the figures, we have a strong proportional joint, but there are several things in connection with it that require to be explained and understood before its design and arrangement can be appreciated.

A practical mechanic, looking at the first joint, would naturally argue something like this: "Yes, the figures as given and worked are all right, but the strength of the plate section has been taken *at the wide pitch only;* the strength at the narrow pitch is never mentioned, and, as the pitch *there* is only $2\frac{3}{16}$in., it is evident the plate section must be very much weaker than at the wide pitch, which is just double—viz., $4\frac{3}{8}$ in.—and as you have already told us that the only way to increase the strength of the plate section was by increasing the pitch of the rivets, it appears that some satisfactory explanation is required in order to clear up this apparent inconsistency."

Now, all this is only what might be expected, but the following explanation may perhaps be sufficient to make it understood.

Look at the figure again, and a little reflection will convince you that before the plate section at the narrow pitch can be separated the shearing strength of the wide pitched rivet must also be overcome, or, to put it another way, we might say that the shearing strength of the wide pitched rivet is to be added to the narrow pitch plate section, and that by doing this it can be proved that the percentage at the narrow pitch, *plus* the shearing strength of wide pitched rivet, is as strong (and, in fact, stronger) than the plate section at the wide pitch.

Another question which would probably crop up is this:—Are we justified in adding the *whole* strength of wide

pitched rivet to the narrow pitch plate section? No, we are not; we can only add one-half of it. Why? Because *there are two pitches to one wide pitched rivet;* therefore the entire shearing strength of this rivet must be equally divided between the two, as can be clearly seen by examining drawing.

To prove what has been said, let us work the whole thing out in plain figures. To repeat: What we want to do is to show that the plate's strength at the narrow pitch, *plus* $\frac{1}{2}$ the shearing strength of the wide pitched rivet, is fully equal to the strength of the plate section at the wide pitch.

The narrow pitch is equal to $\dfrac{4\frac{3}{8}}{2} = 2\frac{3}{16}$ in.

Therefore—Narrow pitch of rivets is $2\frac{3}{16}$ in.
Diameter of rivets is .. $\frac{13}{16}$ in.
Thickness of plates is .. $\frac{5}{8}$ in.

Here $\dfrac{2 \cdot 1875 - \cdot 8125 \times 100}{2 \cdot 1875} = 62 \cdot 8$ per cent. of solid plate,

to which we have to add $\frac{1}{2}$ the strength of a $\frac{13}{16}$ in. rivet *in double shear,* equal in this case to the area of

$$\frac{\cdot 518}{2} = \cdot 259 \text{ of a square inch.}$$

Then $\dfrac{\overset{\text{area.}}{\cdot 259} \times 100 \times 1 \cdot 75 \times 23}{2 \cdot 1875 \times \cdot 625 \times 28} = \cdot 27 \cdot 25$ per cent. of rivet

section to be added to the plate section at the narrow pitch, so that we have—

Percentage of strength at narrow pitch	62·8
$\frac{1}{2}$ the shearing strength of wide pitched rivet ..	27·25
Total strength at narrow pitch + $\frac{1}{2}$ rivet's strength	90·05%

This shows that the narrow pitch, calculated in the above manner, is about $8\frac{1}{2}$ per cent. stronger than the plate section at the wide pitch. But it may be asked, Why should there be a difference of $8\frac{1}{2}$ per cent.; why not arrange so that they

may be the same strength? Simply because in ordinary working practice we cannot arrange the plate and rivet section to correspond exactly; and in joints of this class, where the plates are thin—that is, where the rivet's diameter *exceeds* the plate's thickness—this difference in strength is greatest; but in treble riveted joints of this description, where the rivet's diameter and the plate's thickness approach each other, this difference is much less, as will be shown.

To make the various percentages correspond in this joint would require the introduction of different sized rivets, which is neither desirable nor necessary, but it is satisfactory to know that the fault (if any) is on the right side, and that in all cases the "narrow pitch" is always the stronger.

Whatever small discrepancy there may be in this or any other joint is always checked by the standard rule "that the weakest percentage through plates or rivets invariably forms the basis of the calculation."

In Traill's excellent tables for all kinds of riveted joints, the whole of the proportions, pitches, rivets, etc., are given to three decimal places, but for ordinary boiler-making this is a refinement in accuracy which can scarcely be carried out. Knowing this, it will be observed that in our examples the plate's thickness and rivet's diameter are given to the nearest $\frac{1}{16}$ in., which will commend itself as being more in harmony with practical work.

Then as regards the butt straps for this particular joint, we have to remember that the "narrow pitch" weakens the plate section considerably more than in the ordinary double riveted "butt," where by comparison it will be seen that for the same thickness of plate ($\frac{5}{8}$in.), and in evenly pitched rivets, we get 76 per cent. of plate section, whereas in this joint we only get 62 per cent. This is due to the material difference in the respective pitch of the rivets, which for ordinary riveting is $3\frac{5}{8}$in., whereas in the narrow pitch it is

only $2\frac{3}{16}$in.; therefore, it is evident that in this joint the butt strap must be weakened in the same ratio.

To provide and compensate for this weakness, it becomes necessary to make these particular butt straps stronger, and this is done by making the straps proportionally thicker, hence we have the following rule:—

$$\frac{5 \times T\ (p - d)}{8 \times (p - 2d)} = T_1 \text{ the thickness}$$

when the number of rivets in the inner row is double the number in the outer row. This applies to steel and iron. Applying this to our joints we have:—

$$\frac{5 \times \cdot625 \times (4\cdot375 - \cdot8125)}{8 \times (4\cdot375 - 1\cdot625)} = \cdot5 \text{ or } \tfrac{1}{2}\text{in. thick.}$$

In ordinary riveting the thickness of the butt straps would only be $\frac{3}{8}$in., so that in this case we have to increase the thickness 33 per cent. to bring it up to the standard rule.

Another feature which should be specially noted is that in the ordinary double riveted "butt," we only get two whole rivets in one pitch, whereas in this we have *three*, viz., two whole and two halves in the zig-zag, and three whole rivets in the chain. This is clearly shown by comparing the shaded rivets in Figs. 31 and 32 with Figs. 27 and 28.

Yet another item which requires special consideration is the distance between the rows of rivets, and what we have to provide for is, that each rivet shall have an ample zone or section of metal surrounding it, to take the strain without in any way interfering or encroaching on the zone or section of any other rivet; or to put it another way, we might say that each individual rivet must have a sufficiency of metal around it to take all the strain, and whatever its breadth may be, we must arrange it so as to be entirely *clear* of the zone or section required by any other rivet.

This will be more clearly understood by an examination of Figure 37, which shows the required bands of steel drawn around each rivet, and which is more likely to be impressed on the mind than any elaborate explanation (see pages 168-9).

The Rule for the "Distance Between The Rows" in this joint is as follows (zig-zag riveting):—

$\sqrt{(\frac{11}{20} \times p + d) \times (\frac{1}{20} \times p + d)} = V$, the distance between rows.

Running this out in plain figures we have:—

$\sqrt{(\frac{11}{20} \times 4 \cdot 375 + \cdot 8125) \times (\frac{1}{20} \times 4 \cdot 375 + \cdot 8125)} = 1 \cdot 82$in., or say $1\frac{7}{8}$in., the distance between the rows.

For the "Diagonal Pitch" we have by formula—

$\frac{3}{10} \times p + d = p\text{D}$ which in this case would be:—

$\frac{3}{10} \times 4 \cdot 375 + \cdot 8125 = 2$in. the diagonal pitch.

Take another example, in which the

Wide pitch of rivets	is	$6\frac{3}{4}$in.
Diameter of rivets	,,	$1\frac{3}{8}$in.
Thickness of plate	,,	$1\frac{3}{16}$in.

$$\frac{6 \cdot 75 - 1 \cdot 375 \times 100}{6 \cdot 75} = 79 \cdot 63 \% \text{ of solid plate}$$

$$\frac{\overset{\text{area.}}{1 \cdot 4848} \times \overset{\text{rivets.}}{3} \times 100 \times 1 \cdot 75 \times 23}{6 \cdot 75 \times 1 \cdot 1875 \times 28} = 79 \cdot 88 \% \text{ of rivet section.}$$

Narrow pitch of rivets	is	$3\frac{3}{8}$in. ($\frac{1}{2}$ of $6\frac{3}{4}$)
Diameter of rivets	,,	$1\frac{3}{8}$in.
Thickness of plate	,,	$1\frac{3}{16}$in.

Narrow pitch $\dfrac{3 \cdot 375 - 1 \cdot 375 \times 100}{3 \cdot 375} = 59 \cdot 25 \%$ of solid plate, plus half the shearing strength of the wide pitched rivet, which is $\dfrac{1 \cdot 4848}{2} = \cdot 7424$ of a square inch in area. The percentage to be added to the plate section at narrow pitch is:—

$$\frac{\overset{\text{area.}}{\cdot 7424} \times 100 \times 1 \cdot 75 \times 23}{3 \cdot 375 \times 1 \cdot 1875 \times 28} = 25 \cdot 97 \% \text{ so that we have—}$$

Plate section at narrow pitch	59·25 %
Plus ½ the shearing strength of rivets ..	25·97 %
Total strength at narrow pitch ..	85·22 %
Taking weakest percentage of joint, which is	79·63 %
The narrow pitch is stronger by	5·59 %

Treble Riveted Double Butt Strap Joint.

Each alternate rivet is omitted in the outer and inner rows, as represented in Figures 33 and 34, one being zig-zag and the other chain riveted. As in all the other pairs, both joints are exactly the same strength, only differently arranged.

The explanation given in connection with the double riveted "butt" of the same class applies to this joint also, but the thickness of butt straps in this joint does not require to be increased.

Wide pitch of rivets is .. 5½in.
Diameter of rivets „ .. ⅞in.
Thickness of plate „ .. ¾in.

$$\text{Wide pitch } \frac{5 \cdot 5 - \cdot 875 \times 100}{5 \cdot 5} = 84\ \%\text{ of solid plate.}$$

$$\text{Total rivet section } \frac{\overset{\text{area. rivets.}}{\cdot 6 \times 4} \times 100 \times 1 \cdot 75 \times 23}{5 \cdot 5 \times \cdot 75 \times 28} = 84\ \%\text{ of rivet section.}$$

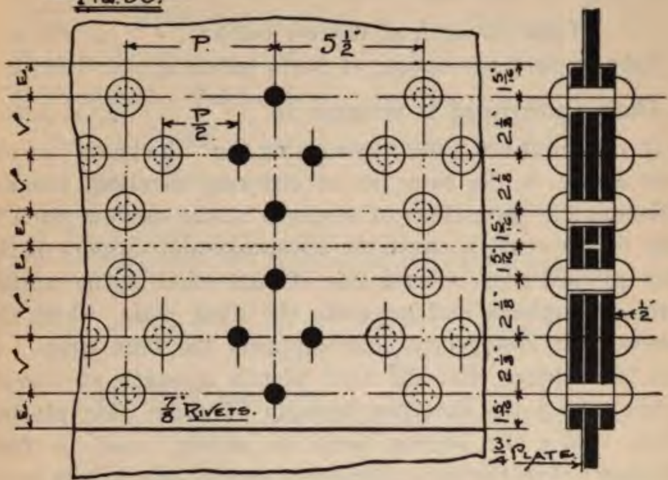
$$\text{Narrow pitch is equal to } \frac{5 \cdot 5}{2} = 2 \cdot 75\text{in.}$$

$$\text{Then } \frac{2 \cdot 75 - \cdot 875 \times 100}{2 \cdot 75} = 68\ \%\text{ of solid plate,}$$

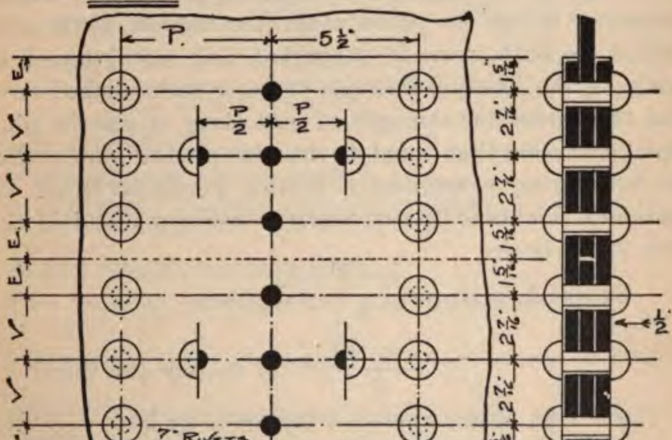
plus half the shearing strength of the wide pitched rivet, which is $\frac{\cdot 6}{2} = \cdot 3$ of a square inch in area. The percentage to be added to the plate section at narrow pitch is

$$\frac{\overset{\text{area.}}{\cdot 3} \times 100 \times 1 \cdot 75 \times 23}{2 \cdot 75 \times \cdot 75 \times 28} = 21\ \%$$

—Fig. 33.—



—Fig 34.—



So that we have: —

Plate section at narrow pitch	68 %
Plus $\frac{1}{2}$ the shearing strength of rivet	21 %
Total strength at narrow pitch	89 %
Take weakest percentage of joint, which is ..	84 %
The narrow pitch is stronger by	5 %

In this joint the "rule" governing the "thickness" of the butt straps is the same as in ordinary riveting, because although the percentage of strength at the narrow pitch is only 68 per cent., we have the advantage and support of the wide pitched rivet on each side of butt—that is, the outside rivet strengthens and supports the shell plate, while the inside rivet strengthens and supports the butt strap. It has been shown that the *shell* plate's strength at narrow pitch, plus $\frac{1}{2}$ the shearing strength of *outer* wide pitched rivet, makes the narrow pitch as strong (and, in fact, stronger) than the wide pitch plate section, the former being 89 and the latter 84 per cent. respectively. And by similar reasoning it can be proved that the narrow pitch plate section in *butt strap* is supported and strengthened by adding $\frac{1}{2}$ the shearing strength of *inner* wide pitched rivet, and this makes the strength of butt strap at narrow pitch equal, and more than equal, to the wide pitched plate section in butt strap, consequently in this particular joint the ordinary thickness of butt strap is sufficient, and in this case would be:—

Double butt straps $\dfrac{5 \times T}{8} = T_1$

$$\frac{5 \times \cdot 75}{8} = \cdot 47 \text{ in., say } \tfrac{1}{2}\text{in. thick.}$$

For V (the distance between the rows) we have

$$\sqrt{(\tfrac{11}{20} \times p + d) \times (\tfrac{1}{20} \times p + d} = V$$

$$(\tfrac{11}{20} \times 5 \cdot 5 + \cdot 875) \times (\tfrac{1}{20} \times 5 \cdot 5 + \cdot 875) = 2\tfrac{1}{8}\text{in.}$$

For p_D (the diagonal pitch), we have

$$\tfrac{3}{10} \times p + d = p_D.$$

In this joint $\frac{3}{10} \times 5 \cdot 5 + \cdot 875 = 2 \cdot 52$in., say $2\frac{1}{2}$in.

Take another example of this joint, where the

Wide pitch of rivets is	..	$6\frac{7}{8}$in.
Diameter of rivets	..	$1\frac{1}{8}$in.
Thickness of plate	..	1in.

Wide pitch plate section:—

$$\frac{6 \cdot 875 - 1 \cdot 125 \times 100}{6 \cdot 875} = 83 \cdot 6 \ \% \text{ of solid plate.}$$

Total rivet section—

$$\frac{\overset{\text{area.}}{1} \times \overset{\text{rivets.}}{4} \times 100 \times 1 \cdot 75 \times 23}{6 \cdot 875 \times 1 \times 28} = 83 \cdot 6 \ \% \text{ rivet section.}$$

$$\text{Narrow pitch is equal to } \frac{6 \cdot 875}{2} = 3 \cdot 4375\text{in.}$$

$$\text{Then } \frac{3 \cdot 4375 - 1 \cdot 125 \times 100}{3 \cdot 4375} = 67 \cdot 27 \ \% \text{ of solid plate,}$$

plus half the shearing strength of the rivet, which is $= \cdot 5$ of a square inch in area. The percentage to be added to the plate section at narrow pitch is

$$\frac{\cdot 5 \times 100 \times 1 \cdot 75 \times 23}{3 \cdot 4375 \times 1 \times 28} = 20 \cdot 9 \ \%,$$

so that we have—

Plate section at narrow pitch ..	.. ..	67·27%
Plus half the shearing strength of rivet	..	20·9
Total strength at narrow pitch	.. ..	88·17
Take weakest percentage of joint, which is		83·6
The narrow pitch is stronger by	.. ..	4·57%

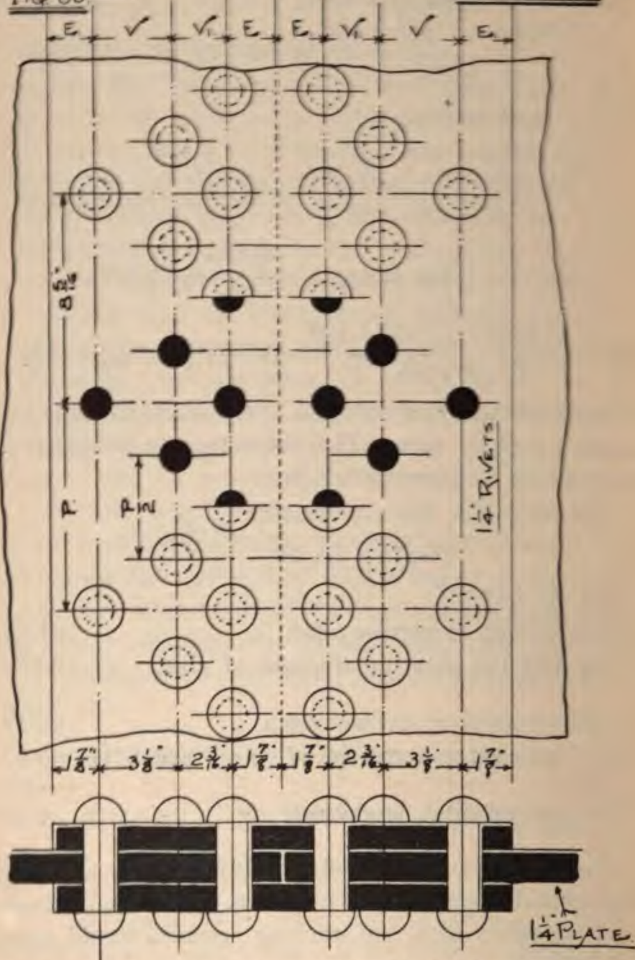
Applying this to a cylindrical 12ft. boiler, assuming the longitudinal seams to be riveted as above, and allowing a factor of safety of 5, the working pressure would come out as follows:—

$$\frac{62720 \times \cdot 836 \times 2}{144 \times 5} = 145\text{lbs. working pressure.}$$

Treble Riveted Double Butt Strap Joint.

In which each alternate rivet is omitted in the outer row. The drawings are for steel plates and steel rivets.

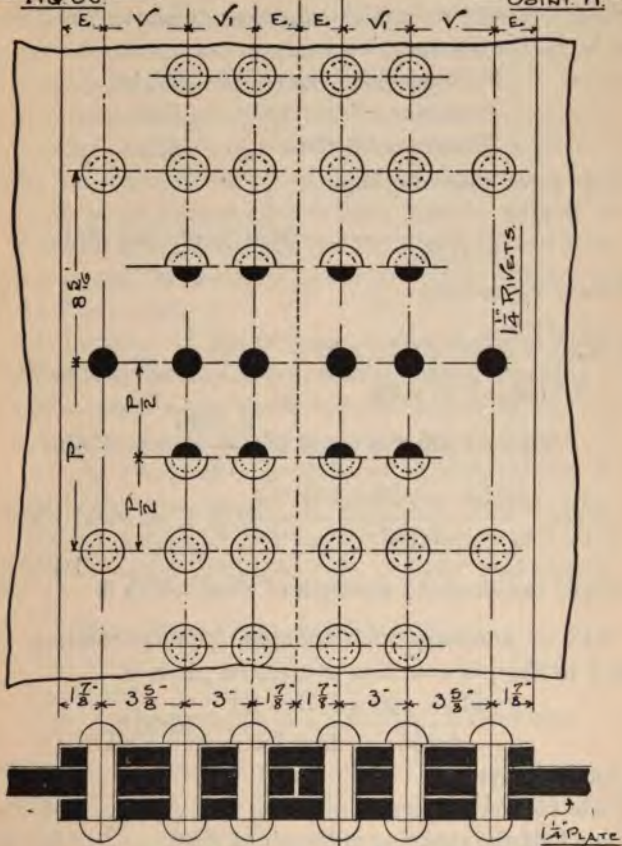
FIG. 35. JOINT. J.



This particular joint is considered the best and most suitable for the present high pressures; it is much more generally used than any other, and is a very good design,

giving us strength and tightness in harmony. Figures 35 and 36 are, as in the other examples, of the same strength; the zig-zag riveting being known as "Joint J," while the chain riveting is recognised as "Joint K."

— FIG. 36 — — JOINT K. —



To ascertain the actual strength of these joints, we use the same formulæ as given in the previous examples, but there are features in connection with the Joints J and K which require to be specially considered and explained. In order that the mechanic may understand and appreciate the arrangement of riveting in this and other joints, Figure

37 has been carefully prepared, showing clearly what the plates, rivets, and butt straps have to do, and a quiet study of the same will do more towards giving the engineer or boiler-maker a good grip of the subject than if he were to attend a course of lectures (see pages 168-9).

First, let us run out the figures, as shown in Joint J, that is, in the zig-zag riveting.

Wide pitch of rivet	..	$8\frac{5}{16}$in.
Diameter of rivet	..	$1\frac{1}{4}$in.
Thickness of plate	..	$1\frac{1}{4}$in.

Wide pitch plate section—

$$\frac{8{\cdot}3125 - 1{\cdot}25 \times 100}{8{\cdot}3125} = 84{\cdot}9\ \%\ \text{of solid plate.}$$

Total rivet section—

$$\frac{\overset{\text{area.}}{1{\cdot}227} \times \overset{\text{rivets.}}{5} \times 100 \times 1{\cdot}75 \times 23}{8{\cdot}3125 \times 1{\cdot}25 \times 28} = 84{\cdot}7\ \%\ \text{of rivet section.}$$

Narrow pitch is equal to $\frac{8{\cdot}3125}{2} = 4{\cdot}156$in.

Then $\frac{4{\cdot}156 - 1{\cdot}25 \times 100}{4{\cdot}156} = 69{\cdot}9\ \%$ of solid plate,

plus half the shearing strength of rivet which is $\frac{1{\cdot}227}{2}$ equal to $\cdot 613$ of a square inch in area. The percentage to be added to the plate section at narrow pitch is

$$\frac{\cdot 613 \times 100 \times 1{\cdot}75 \times 23}{4{\cdot}156 \times 1{\cdot}25 \times 28} = 16{\cdot}91\%$$

so that we have

Plate section at narrow pitch ..	.. ..	69·9%
Plus half the shearing strength of rivet	..	16·91%
Total strength at narrow pitch	.. ..	86·81%
Take weakest percentage of joint, which is	..	84·70%
The narrow pitch is stronger by	.. ..	2·11%

Respecting the butt straps in this J joint, it will be observed that the arrangement of riveting is quite different

from the treble riveted butt shown in Fig. 23, where narrow pitch is in the centre with a wide pitched outer and inner row, and where the outer wide rivet strengthens the shell plate and the inner wide rivet strengthens the butt strap.

In the J joints, as in the other examples, the *outer* rivets strengthen shell plate at narrow pitch, but there is nothing to support or strengthen narrow pitch in butt strap, consequently whatever deficiency or difference there is between the narrow and wide pitch has to be made up or compensated by increasing the butt strap's thickness over and above that required for ordinary evenly pitched riveting.

Our wide pitch plate section is 84·7, and the narrow pitch only 69·9 per cent.; therefore the strap's thickness must be increased so that its strength at narrow pitch will also come out at 84·7 per cent.

An examination of the Figures J and K shows that the arrangement of riveting is such that we have 5 rivets in each main pitch—four whole rivets and two halves in the zig-zag (J), and three whole rivets and four halves in the chain (K). All the rivets in each pitch are shaded, clearly showing what each rivet has to do, and this method may, as before stated, materially assist practical men in getting hold of what is in many instances only vaguely understood.

Butt Strap Formula for J Joint.

$$\frac{5 \times T \times (p - d)}{8 \times (p - 2d)} = T_1, \text{ the strap's thickness.}$$

$$\frac{5 \times 1{\cdot}25 \times (8{\cdot}3125 - 1{\cdot}25)}{8 \times (8{\cdot}3125 - 2{\cdot}5)} = {\cdot}94, \text{ or, say, } \tfrac{15}{16}\text{in. thick}$$

For the distance between the rows the pitching of the rivets requires special consideration. It will be noticed that in the Figure 35 V (the distance between the *outer* and middle row) equals $3\frac{1}{8}$in., while V_1 (the distance between the *inner* and middle row) equals $2\frac{3}{16}$in. only, resulting, of course, in a corresponding difference in the diagonal pitches. There is a very natural reason for this, as will be shown.

The "formula" for finding V (the distance between the outer and middle rows of rivets) is:—

$$\sqrt{\left(\frac{11}{20} \times p + d\right) \times \left(\frac{1}{20} \times p + d\right)} = \text{V., distance between}$$

outer and middle rows.

$$\sqrt{\left(\frac{11}{20} \times 8 \cdot 3125 + 1 \cdot 25\right) \times \left(\frac{1}{20} \times 8 \cdot 3125 + 1 \cdot 25\right)} = 3\tfrac{1}{8}\text{in.}$$

For finding V_1 (the distance between the two inside rows of rivets) we have

$$\sqrt{\frac{(11 \times p + 8 \times d) \times (p + 8 \times d)}{20}} = V_1, \text{ the distance between}$$

the two inside rows.

$$\sqrt{\frac{(11 \times 8 \cdot 3125 + 8 \times 1 \cdot 25) \times (8 \cdot 3125 + 8 \times 1 \cdot 25)}{20}} = 2\tfrac{3}{16}\text{in.}$$

For the diagonal pitch (inner row) we have

$$\frac{3 \times p + 4 \times d}{10} = \text{P}_\text{D}, \text{ diagonal pitch of inner row.}$$

$$\frac{3 \times 8 \cdot 3125 + 4 \times 1 \cdot 25}{10} = 3\text{in., diagonal pitch of inner row.}$$

For diagonal pitch (outer row)—

$$\tfrac{3}{10} \times p + d = p_\text{D} = \text{diagonal pitch in outer row.}$$

$$\tfrac{3}{10} \times 8 \cdot 3125 + 1 \cdot 25 = 3\tfrac{3}{4}\text{in., diagonal pitch in outer row.}$$

In the chain riveted joint (K) the percentages of strength through plates and rivets are calculated in the same way, and give the same result as in joint J, but the distances between the rows of rivets are different.

For finding V (the distance between the outer and middle row of rivets) the formula is

$$\sqrt{\frac{(11 \times p + 4 \times d) \times (p + 4 \times d)}{10}} = \text{V.}$$

$$\sqrt{\frac{(11 \times 8 \cdot 3125 + 4 \times 1 \cdot 25) \times (8 \cdot 3125 + 4 \times 1 \cdot 25)}{10}} = 3\tfrac{5}{8}\text{in.}$$

To find V_1 (the distance between the two inside rows of rivets) the rule is

$$\frac{4 \times d + 1}{2} = V_1$$

$$\frac{4 \times 1{\cdot}25 + 1}{2} = 3\text{in.}$$

Figure 37 (on pages 168-9) is a representation of a J joint. One part shows the shell plates and rivets, with the butt strap removed, while the other shows the joint with the butt strap in position. It will be observed that around each rivet is drawn a band of steel of such breadth and section as will be at least equal to the shearing strength of the rivet, and it will also be noted that the joint is designed in such a manner that each zone or band can be of sufficient breadth without touching, encroaching, or interfering with the zone or band required for any other rivets. This is plainly seen in both the shell plates and butt straps.

In the shell plates, the section shown around each rivet is fully equal to the rivets' shearing strength, and in the butt strap, the combined section (one on each side) is also equal, and more than equal to the shearing strength of the rivets. By having such a combination, we establish a relationship between shell plates, butt straps and rivets the result of which is a strong proportional reliable structure. From what has been said, it is evident that all the material outside the dotted steel band is practically useless, so far as taking any of the strain is concerned; and in this sense, the sectional area in the steel bands do all the work; the material outside of them simply holds the water in. Any increase in the breadth and section of the bands would be of no benefit, because the amount of section required around each rivet is and must always be governed by the rivet's shearing strength. If we were to test this boiler joint against a structure made of steel bands and rivets only, as shown, the result would be practically the same.

The particulars of Figure 37 are as follows:—

T	= The thickness of plates	$1\frac{5}{16}$in.
d	= the diameter of rivets	$1\frac{5}{16}$in.
p	= the wide pitch of rivets	$8\frac{3}{4}$in.
$\frac{p}{2}$	= narrow pitch of rivets	$4\frac{3}{8}$in.
E	= from centre of rivets to edge of plate ..	2in.
V	= distance between outer and middle row of rivets	$3\frac{1}{4}$in.
V^1	= distance between the two inner rows of rivets	$2\frac{1}{4}$in.
p_D	= diagonal pitch between outer and middle row	4in.
P_D	= diagonal pitch between inner and middle row	$3\frac{1}{8}$in.
T_1	= thickness of butt straps	1in.

Strength of joint when so proportioned is 85% of solid plate. Applying this system of riveting to a steel boiler 14 feet inside diameter, with a factor of safety of 5, we would get:—

$$\frac{62720 \times \cdot 85 \times (1 \cdot 3125 \times 2)}{168 \times 5} = 165\text{lbs. } W.P.$$

Another very important feature in connection with this joint is the following:—

It will be noticed that there is a material difference in the distance between the rows of rivets and also the diagonal pitches. The distance between the two inner rows is $2\frac{1}{4}$in., the diagonal pitch being $3\frac{1}{8}$in., but the distance from middle to outer row is $3\frac{1}{4}$in., while the diagonal pitch is 4in. What is the reason of this? A quiet examination of the Figure (where the butt strap is removed) will show that to get the required breadth of "band" around the rivet A, without encroaching on the bands required for rivets B and C, it is absolutely necessary to increase the diagonal pitch so that every rivet shall have a sufficient and independent zone of its own. If the diagonal pitch between the outer and middle rows were the same as between the inner and middle rows, viz., $3\frac{1}{8}$in., the rivet A would be $\frac{7}{8}$ths of an inch further in, and we would have the three rivets A, B, and C, all pulling on the *same zone of metal*, which, of course, would

make a ridiculous arrangement, hence it is that the design of the shell, plates, rivets, and butt straps must be as shown.

Referring to butt straps generally, the rule given that their combined thickness must be 25 per cent. in excess of the plates they cover is one which, to a practical mind, will not appear logical or clear; but the results of actual testing, the possibility of leverage being developed if the strains are not equally distributed throughout the entire bearing surface of the plate and rivets, the strains in shells not being straight, but curved, and the absolute necessity (in J and K joints especially) of having the butt straps thick enough to bear caulking without the possibility of any "spring" between the rivets are sufficient reasons for adopting the rule.

To show that the butt straps in joint J (Fig 35) comply with the above:

Wide pitch of rivets ..	..	8·317in.
Narrow pitch of rivets	..	4·158in.
Diameter of rivets ..	..	1·25
Thickness of plate ..	..	1·25
Thickness of butt strap	..	$\cdot 94 \times 2 = 1 \cdot 88$in.

Butt Straps—Material left in Narrow Pitches.

4·158in.	the narrow pitch
1·25	the diameter of rivet
2·908	surface inches in one narrow pitch
2	pitches (one on each side)
5·816	surface inches in the two narrow pitches
1·88	double thickness of butt strap
46528	
46528	
5816	
10·93408	square inches of metal in the double strap at the narrow pitch.

Shell Plates—Material left in Wide Pitch.

8·317in.	the wide pitch
1·25	the diameter of rivet
7·067	surface inches left in wide pitch
1·25	thickness of plate
35335	
14134	
7067	
8·83375	square inches of metal in the wide pitch
1·25	butt straps must have 25% more material than shell plates
4416875	
1766750	
883375	
11·0421875	square inches required in butt strap.

And as we have got 10·93 sq. in., it practically proves our work to be correct and that the rule has been complied with.

This completes the riveting, and, from the number of joints of various designs having different and distinct arrangements of riveting, it will be apparent that this is not the least important part of our subject; but as each joint has been separately treated, illustrated, and explained, both by formulæ and plain figures, there will be no serious difficulty to the engineer or boiler-maker in understanding, appreciating, and applying the principles of riveting to every-day work.

The following tables relating to riveting in general will be found useful and easily applied:—

No. 1

Table of Pitches in Terms of Rivets Diameter.

For	55%	plate	left	between holes,	pitch =	diameter ×	2·222
,,	56	,,	,,	,,	,,	,,	2·273
,,	57	,,	,,	,,	,,	,,	2·326
,,	58	,,	,,	,,	,,	,,	2·381
,,	59	,,	,,	,,	,,	,,	2·439
,,	60	,,	,,	,,	,,	,,	2·5
,,	61	,,	,,	,,	,,	,,	2·564
,,	62	,,	,,	,,	,,	,,	2·631
,,	63	,,	,,	,,	,,	,,	2·702
,,	64	,,	,,	,,	,,	,,	2·777
,,	65	,,	,,	,,	,,	,,	2·857
,,	66	,,	,,	,,	,,	,,	2·941
,,	67	,,	,,	,,	,,	,,	3·03
,,	68	,,	,,	,,	,,	,,	3·125
,,	69	,,	,,	,,	,,	,,	3·225
,,	70	,,	,,	,,	,,	,,	3·333
,,	71	,,	,,	,,	,,	,,	3·448
,,	72	,,	,,	,,	,,	,,	3·571
,,	73	,,	,,	,,	,,	,,	3·703
,,	74	,,	,,	,,	,,	,,	3·846
,,	75	,,	,,	,,	,,	,,	4·
,,	76	,,	,,	,,	,,	,,	4·166
,,	77	,,	,,	,,	,,	,,	4·347
,,	78	,,	,,	,,	,,	,,	4·545
,,	79	,,	,,	,,	,,	,,	4·762
,,	80	,,	,,	,,	,,	,,	5·

NOTE.

The pitch of rivets may be increased directly as the thickness of plate, viz.:—If a 2½in. pitch be right for a ½in. plate, then 5in. will not be too much for a 1in. plate.

No. 4.

Treble Riveted Lap Joints.

Table of Diameters in Terms of Thickness of Plate.

Actual % of strength of joint to solid plate.	Where the Shearing and Tensile Strengths are equal.	Where $\frac{\text{Tensile}}{\text{Shearing}} = \frac{28}{23}$ (for Steel Riveting.)
For 70%	Diameter = thickness × ·99	Diameter = thickness × 1·205
,, 71	,, ,, 1·039	,, ,, 1·264
,, 72	,, ,, 1·091	,, ,, 1·328
,, 73	,, ,, 1·147	,, ,, 1·396
,, 74	,, ,, 1·208	,, ,, 1·47
,, 75	,, ,, 1·273	,, ,, 1·55
,, 76	,, ,, 1·344	,, ,, 1·635
,, 77	,, ,, 1·421	,, ,, 1·73
,, 78	,, ,, 1·505	,, ,, 1·83

No. 3.

Double Riveted Lap Joints.

Table of Diameters in terms of thickness of Plate for different percentages of strength of joint.

Actual % of strength of joint to solid plate.	Where the Tensile and Shearing Strengths are equal.	Where $\frac{\text{Tensile}}{\text{Shearing}} = \frac{28}{23}$ (for Steel Riveting)
For 68%	Diameter = thickness × 1·352	Diameter = thickness × 1·646
„ 69	„ „ 1·416	„ „ 1·724
„ 70	„ „ 1·485	„ „ 1·807
„ 71	„ „ 1·558	„ „ 1·896
„ 72	„ „ 1·636	„ „ 1·992
„ 73	„ „ 1·722	„ „ 2·095
„ 74	„ „ 1·811	„ „ 2·205
„ 75	„ „ 1·909	„ „ 2·323

No. 6.

Treble Riveted Double Butt Strap Joints.

Table of Diameters in terms of thickness of Plate where 1·75 is taken as value for Double Shear.

Actual % of strength of joint to solid plate.	Where the Tensile and Shearing Strengths are equal.	Where $\frac{\text{Tensile}}{\text{Shearing}} = \frac{28}{23}$ (Steel Riveting).
For 76%	Diameter = thickness × ·768	Diameter = thickness × ·935
,, 77	,, ,, ·812	,, ,, ·988
,, 78	,, ,, ·860	,, ,, 1·050
,, 79	,, ,, ·912	,, ,, 1·110
,, 80	,, ,, ·970	,, ,, 1·180

The diameter of rivet must always be found before determining the pitch, by assuming desirable percentages of strength at joint, both for plate and rivets. The diameter being round, the pitch is arrived at by making use of the same percentage of plate section as was assumed.

Safety Valves.

Safety valves should be so constructed that under all possible conditions of working they should be capable of getting rid of the steam as fast as it is generated, or, at all events, without any undue accumulation of pressure over and above that at which the boiler is authorised to work.

Practical experience and actual test have determined the relative proportions for obtaining the highest efficiency, and modern valves, properly designed and constructed, leave little to be desired, the results obtained being highly satisfactory.

Generally valves may be loaded in three different ways: 1st, by dead weight; 2nd, by levers; and 3rd, by direct springs.

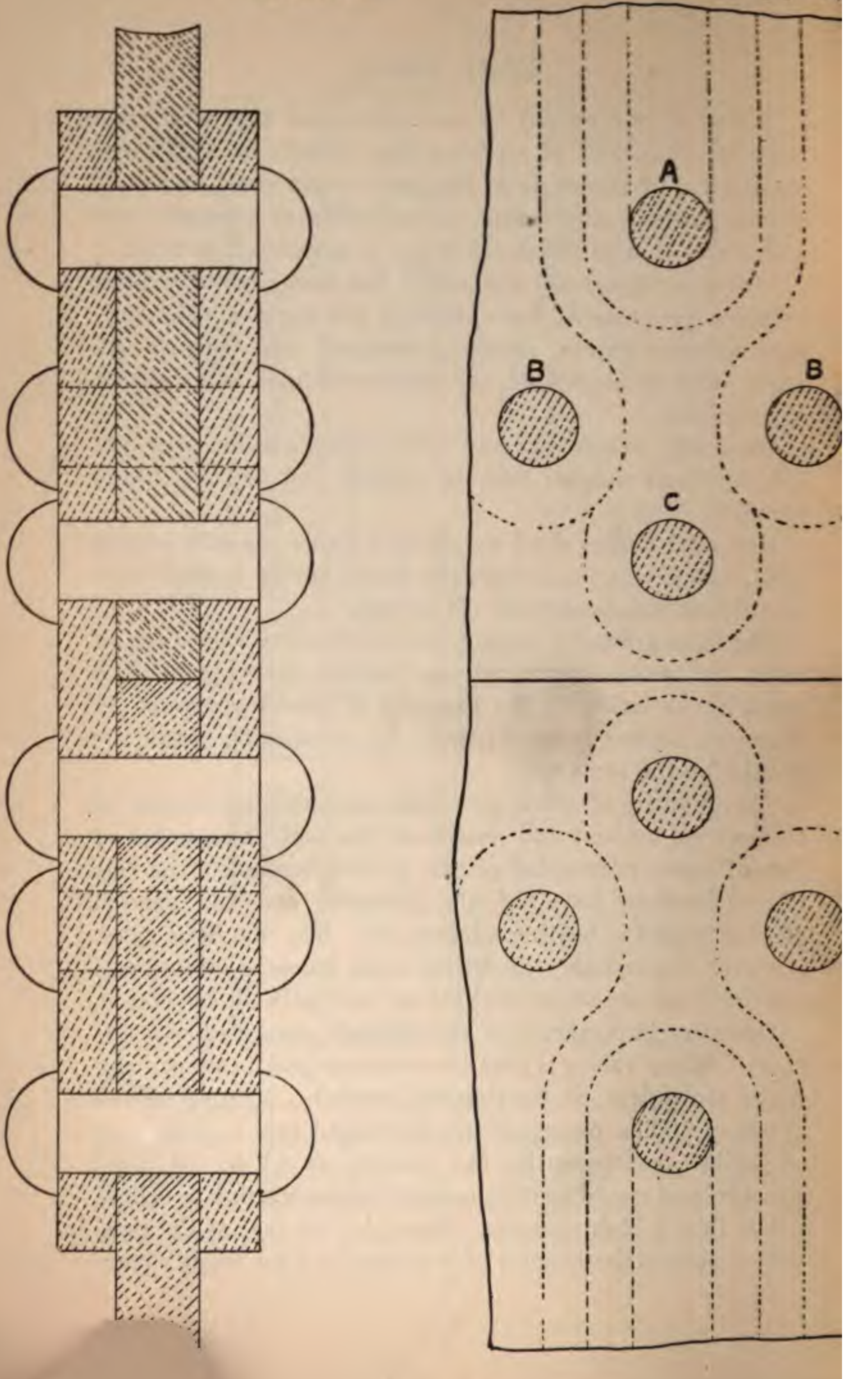
For land boilers dead weight and levers are still largely used, but in marine boilers the direct spring loaded valve has almost superseded the old system.

The rules given for finding the size of safety valves are all more or less approximations, because there is often a material variation in the quantity of steam generated in different boilers, whose powers by calculation would or should be the same.

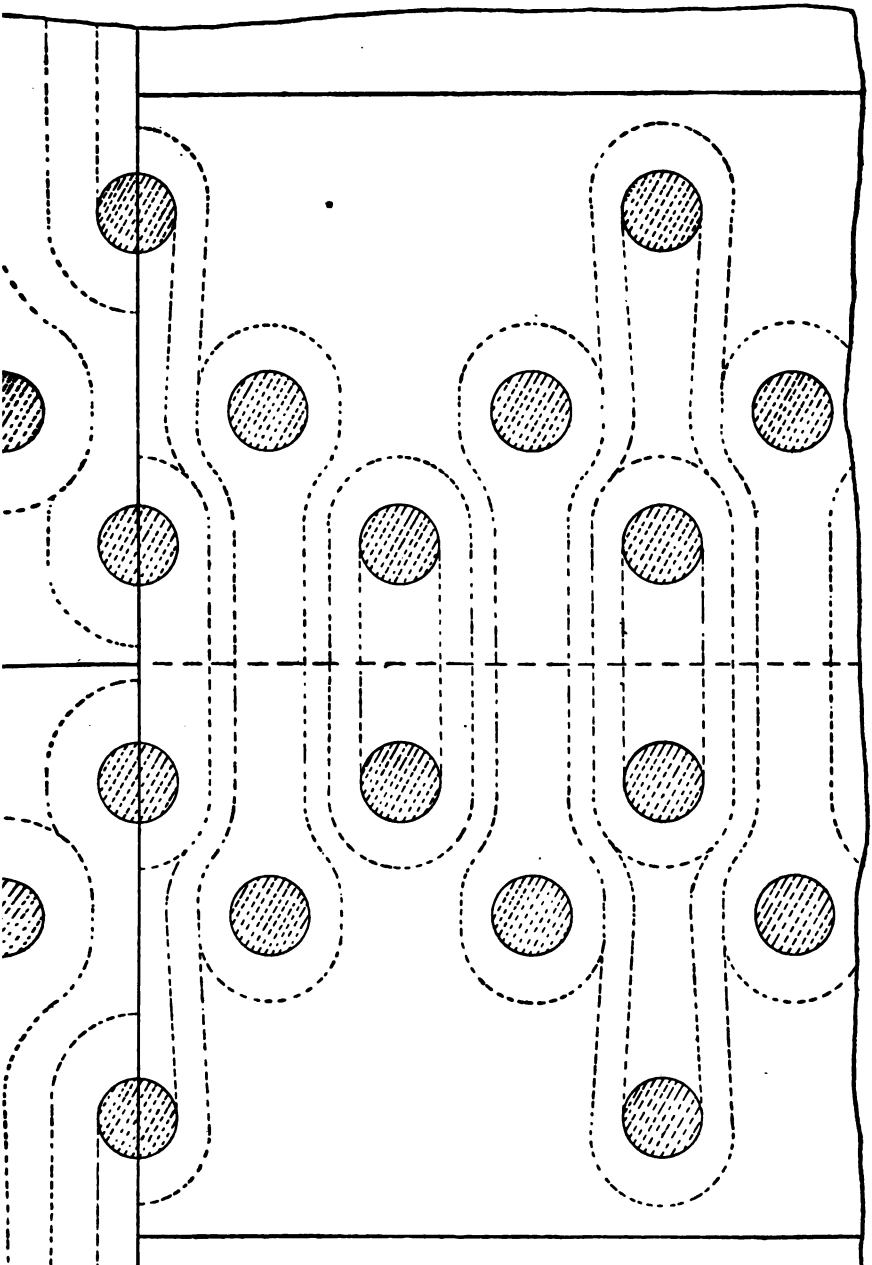
The quantity of steam generated depends on a number of things: the amount and quality of the coal, the number of lbs. of water evaporated per lb. of fuel consumed, the rate of combustion, force of the draught, and the general efficiency of the heating surface, etc. The standard rules, however, for ordinary conditions make ample allowance and provision for the above, and can be used with all confidence.

One very important item immediately connected with the size of safety valves is that the area required to relieve any boiler depends upon the pressure carried. A cubic foot of steam at 100lbs. per sq. in. would expand into 10 cubic feet at 10lbs.; consequently, the volume would be 10 times greater, and would require a much larger opening to relieve a low than a high pressure. Hence is that that as the pressure increases the amount of area required for every square

—Fig: 37. — View of Buttst



D. — View Shewing Buttstrap. —



foot of grate becomes less, and from various elaborate and exact experiments special tables have been prepared by the Board of Trade for pressures from 15 to 200lbs., by which the required area for any sized valve can be easily found. (See Rules.) The foundation of the calculation is the number of square feet of grate bar surface, but this alone will not in exceptional cases be sufficient.

For ordinary or natural draught it is right enough, but if forced draught be used the rate of combustion and the efficiency of evaporation must be considered and allowed for. If, for example, with forced draught we burn twice the coal and evaporate twice the quantity of water (*in the same time*) that we can with natural draught, then the present rules will not apply, and we must increase the valve area in proportion.

Loading dead weight valves simply means that if the area of valve multiplied by the pressure equals 1000lbs., then 1000lbs. weight attached direct to top or bottom spindle, or both (including weight of spindle and valve), will balance the boiler pressure.

Such valves, though somewhat cumbersome, are very suitable for land boilers, as they possess two essential qualities for this purpose, viz., simplicity and reliability.

Figure 38 shows an ordinary lever valve proportioned on the "Salter's balance principle"—that is, if the boiler pressure be 60lbs. per sq. in., then 60lbs. weight on the lever end will exactly balance that pressure.

This is done by making the area of valve multiplied by the distance from centre of fulcrum pin to centre of valve equal to the lever's whole length.

To illustrate—in the Figure (38) the

Diameter of valve is .. .. 3in.

Distance from fulcrum to valve is $3\frac{3}{4}$in.

The area of a 3in. valve is 7 square inches.

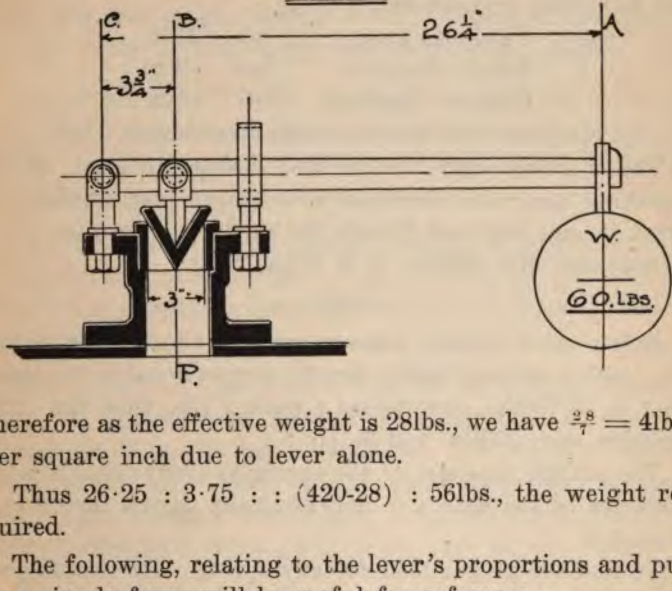
$\therefore 7 \times 3{\cdot}75 = 26{\cdot}25$, the whole length of lever.

Say the pressure carried is 60lbs. per sq. in. Find weight required on lever end.

Here it is evident that the long arm should bear the same proportion to the short arm as the power does to the weight. Neglecting for a moment the effective weight of lever and valve, we have 26·25 : 3·75 : : 420 : 60lbs., the weight required, proving that 60lbs. on boiler is exactly balanced by 60lbs. on lever end.

Assume the effective weight of lever and valve to be 28lbs., then the required weight would be reduced from 60 to 56lbs., because as the power is multiplied 7 times, and as the area of valve is 7 square inches, one pound on lever end will balance the 7 square inches in valve, at 1lb. pressure;

—Fig. 38.—



therefore as the effective weight is 28lbs., we have $\frac{28}{7} = 4$lbs. per square inch due to lever alone.

Thus 26·25 : 3·75 : : (420-28) : 56lbs., the weight required.

The following, relating to the lever's proportions and put in a simple form, will be useful for reference:—

In Figure 38, we have by proportion—

AC	:	BC	: :	P	:	W	the weight	
$26\frac{1}{4}$	:	$3\frac{3}{4}$	: :	420—28	:	56	„	
W	:	P	: :	BC	:	AC	whole length of lever	
56	:	420—28	: :	$3\frac{3}{4}$	:	$26\frac{1}{4}$	„	

P	:	W	: :	AC	:	BC	distance from fulcrum [to valve.
420—28	:	56	: :	$26\frac{1}{4}$	:	$3\frac{3}{4}$	,,
BC	:	AC	: :	W	:	P	
$3\frac{3}{4}$	:	$26\frac{1}{4}$	: :	56	:	420—28	the steam power

And power $\div$ total pressure $=$ area of valve.

Thus $\frac{420}{60} = 7$ square inches in valve.

*And $\sqrt{7} \times 1\cdot128 = 3$ inches, the diameter of valve.

Spring Safety Valves.

The standard spring from which Adams' springs are calculated was made of $\frac{1}{2}$in. square bar steel, and was of the following proportions:—

Sectional area	=	·25in.
Inside diameter	=	2in.
Outside diameter	=	3in.

13 complete coils and the ends, total length, $11\frac{1}{2}$in.

The working load was 600lbs., being $\frac{1}{8}$th part of its breaking load when hardened to a temper just sufficient to break it, and this load deflects the spring 1in., so that when compressed with 600lbs., it is $10\frac{1}{2}$in. long.

Note.

Safety valve springs, when properly made and tempered, are (within certain limits) directly proportional to the load; that is, if 100lbs. compresses a spring $\frac{1}{4}$in., then the compression with 200lbs. will be $\frac{1}{2}$in.

The outside diameter of the spring is always equal to the diameter of the steel $\times$ 6, and generally equals the valve's diameter.

To find the proper load for any spring. By simple proportion we have:—

Section of $\frac{1}{2}$in. square bar : 600lbs. : : section of desired spring : intended load.

*The "constant" 1·128 is used because the side of any square multiplied by 1·128 gives the diameter of a circle of equal area. A circle to be equal to one square inch would be 1·128 inches in diameter.

Examples.

Let A = Section of ½in. spring = ·25 of a sq. in. in area
„ L = Load on it; viz., 600 lbs.
„ S = Section of desired spring
„ L' = Load you want to find.

Then $\frac{S \times L}{A}$ = L', or taking it practically, suppose we want to find the proper load to put upon a ¾in. square spring, we would have this proportion :—

area.		load.		area.		load.
·25	:	600	: :	·56	:	1344

Expressing the above in words, we would say :—
That the area of ½in. spring bears the same proportion to 600lbs. as the area of a ¾in. spring bears to 1344lbs.

The proportions of a ¾in. spring by geometrical construction would be 13 complete coils and the ends. Total length, 17¼in. Deflection of spring, 1½in., by which the "lift" of valve is proportional to the diameter.

For example, take two valves—one 3in. in diameter, with a ½in. spring, its deflection = 1in. and the lift ⅛in.; let the other valve be 4½in. in diameter, with a ¾in. spring, its deflection would be 1½in. Then

3in. : ⅛in. : : 4½in. : $\frac{3}{16}$ in. the lift of a 4½in. valve, and the deflection of springs so constructed is directly proportional to the inside diameter; for the ½in. spring deflected 1in. with 600lbs. on it = half its inside diameter; and the ¾in. spring deflected 1½in. = half its inside diameter, when loaded by 1344lbs.; and so on with all other sizes.

Modern spring valves are so correctly made that they are absolutely accurate as a register of force between the limit of 1lb. and the load that produces permanent set.

The Board of Trade Rules give all the necessary formulæ for finding the working pressures, etc., to be allowed on different sized springs—(see Rules); and the following is

an example, when the load is known, how to find the section of the steel:—

Rule.

Multiply the load on the valve by the helical diameter (centre to centre of section) of spring, and divide by 11000, then extract the cube root, and the quotient is the side of square of the steel in parts of an inch.

$$\sqrt[3]{\frac{L \times D}{C}} = \text{required diameter or side of square.}$$

Where L = load in lbs.

D = helical diameter (centre to centre of steel).

C = 11000 for square steel.

C = 8000 for round steel.

Let D = 3·75in.

L = 1344lbs.

C = 11000.

$$\text{Then } \sqrt[3]{\frac{1344 \times 3{\cdot}75}{11000}} = {\cdot}77\text{in. the side of square.}$$

Copper Steam Pipes.

The strength of copper in tension is variable, but copper suitable for steam pipes may be averaged as being equal to 30000lbs. (13·4 tons) per square inch. The strength of a well brazed joint is sometimes quite as strong as the solid plate, but as a general rule the "brazing" is more or less weaker. One important point in connection with copper steam pipes is that they lose a considerable percentage of their strength at a comparatively low temperature.

In iron and steel it has been repeatedly proved that up to about 600° Fah. the strength actually, though slightly, increases. Not so with copper, for at modern pressures, from 150 to 200lbs. per sq. in., it has been demonstrated that the temperature of the steam above, say, 380° Fah. reduces the strength of copper pipes from 20 to 25 per cent., and therefore in calculating the working pressure due allowance

has to be made for this, and also for brazing, which as often as not is 10 or 15 per cent. less in strength than the solid plate. For present pressures up to, say, 160lbs., a simple reliable rule is to take the copper at 24000lbs. (20 per cent. below its initial strength), and allow a factor of safety of 10. Then

$$\frac{2400 \times (\text{thickness in in.} \times 2)}{\text{Inside diameter in in.}} = \text{Working pressure.}$$

For example: An 8in. steam pipe is $\frac{1}{4}$in. thick. At what pressure would we work it?

$$\text{Here } \frac{2400 \times \cdot 5}{8} = 150, \text{ the working pressure.}$$

From the above it will be seen that reducing the copper from 30000 to 24000 makes due allowance for the loss due to the increase of temperature and the margin of 10 per cent. for brazing.

The Board of Trade "formula" for well-made copper pipes when the joints are brazed is—

$$\frac{6000 \times (T - \frac{1}{16})}{D} = \text{Working pressure.}$$

Where T = thickness in inches

D = inside diameter in inches.

When the pipes are solid drawn and not more than 8in. in diameter, substitute $\frac{1}{32}$ for $\frac{1}{16}$.

This rule gives a stronger pipe, and at first sight appears somewhat inconsistent, especially for thin pipes.

If, for instance, the thickness was $\frac{1}{8}$in., by taking $\frac{1}{16}$ off the thickness, the strength would be reduced 50 per cent., which is too much; but as the thickness increases, the amount of reduction will, of course, be less, and no doubt this $\frac{1}{16}$ is introduced to make it thoroughly reliable and provide for "wear," the thinning of the copper at bends, loss through increased temperature, etc., so that, everything considered, the fault (if any) is on the right side and gives additional strength and security.

Wrought Iron Pipes.

The internal pressure to be allowed on wrought iron pipes, made of good material, which are lap-welded and thoroughly sound, may be determined by the following Board of Trade rule, provided the thickness is not less than $\frac{1}{4}$in. (same notation) :—

$$\frac{6000 \times T}{D} = \text{Working pressure.}$$

The working pressure on our 8in. pipe, if made of iron, would be

$$\frac{6000 \times \cdot 25}{8} = 187\text{lbs. per square inch.}$$

Cast Iron Steam Pipes.

As cast iron in tension is about $\frac{1}{3}$rd the strength of wrought iron, the most natural thing to do would be to reduce the constant for wrought iron two-thirds, viz., from 6000 to 2000. This would be all right provided cast and wrought iron were equally reliable, but they are not, because we can never depend on any cast iron pipe having a uniform section; the "core," as a rule, can scarcely be kept in the exact centre, and generally shifts more or less. We have, therefore, to allow for this inequality of section, and also for corrosion, hence the desirability of reducing the 2000 to 1500, and this will give a fair average rule for pipes not less than 6in. in diameter. Cast iron steam pipes for extreme pressures can scarcely be recommended; they are not allowed in marine practice and for land purposes, copper or iron are much more reliable.

Size of Steam Pipes.

This depends on several things, viz., size of cylinder, point of cut off, capacity of receiver, speed of piston, etc., but is principally regulated and determined by the possible velocity that can be allowed to steam without developing an undue amount of friction.

From experiment it has been shown that when the average steam pipe, with moderate bends, has a cross sectional area which will allow a speed of 133 feet per second, or say 8000 feet per minute, then there is little or no loss of initial pressure (due to frictional flow) between the boiler and the engine, and a good general rule, approximately correct, might be expressed thus:—

$$\sqrt{\frac{\text{Dia. of cyl.}^2 \times \text{Piston speed in ft. per min.}}{\text{Velocity of steam per minute.}}} = \text{dia. of pipe.}$$

Example.

Let cylinder	be	30in. in diameter
„ Piston speed	„	600 feet per minute
„ velocity of steam	„	8000 „ „ „

Then $\sqrt{\frac{30^2 \times 600}{8000}} = 8{\cdot}2$in., the diameter of pipe.

Heating Feed Water.

Our present practice of feeding boilers has very little to commend it, and there is ample room for improvement.

It is both important and necessary that in all boilers the "feed" should, if possible, be heated up to the temperature of the steam *before* being discharged.

This may be done in various ways; the most simple and effective method being to pump the feed into an internal pipe in the steam space, this pipe being of such length and surface as will allow and ensure every particle of feed water being heated up to the temperature of the steam before being discharged. By this method we secure two great advantages:

1st. Every particle of water in the boiler has an equal chance of generating steam, and is not bustled and handicapped as in ordinary practice, where even in the most modern feed-heaters there is often a temperature difference of 160° to 200° between the water going in and the water that is in. Take Weir's feed-heater, which usually delivers

the feed at 200° Fah., and assume the boiler is working at 160lbs. The temperature of the steam is 370°—a difference of 170°—which must of necessity create a considerable commotion and disturbance amongst the particles and injuriously affect their individual efficiency for generating steam.

2nd. In all modern appliances for feed-heating, the steam is taken *out* of the boiler, passed through pipes, cocks, coils, etc.; then forced through other pipes and check-valves into boilers. Now, it is very apparent that this cannot be done without a certain percentage of loss through friction, condensation, etc. Therefore by discharging the "feed" direct into a pipe in the steam-space there must be a distinct gain, besides being cheap, simple and efficient.

There is one thing, however, that requires special consideration when the feed is heated in the steam space. Special care must be taken that the cold water does not suddenly come into contact with the internal pipe *when it is full of steam,* because if it does, the sudden condensation of the steam at one point causes a "water-hammer" action and excessive vibration, resulting in leaky joints and damaged pipes. With proper precautions, however, this system of feeding gives no trouble, and practical experience has shown that its application has been exceptionally economical.

Another important item, which for many years has been neglected, is the filtering and purification of all feed water. With present pressures in Triple and Quadruple Engines this is especially a vital point, as practical working has proved that even with the best mineral oils, whose specific gravity is say 10 per cent. less than water, yet, after passing engines and condensers, the chemical change is such that when it gets into the boiler it becomes as heavy, and in some cases heavier than water, falls down and settles on the internal surfaces, more especially on furnace crowns, and, being an almost perfect non-conductor, is in many instances the cause of furnace-plates becoming over-heated. This is a practical mechanical difficulty only, and can be

successfully overcome by purifying the water *before* it gets into the boiler.

The best modern practice is beginning to realise the absolute necessity of making special provision for water purification, and a number of patents have been taken out which have been more or less successful.

One of the most simple and effective schemes is to make a plain iron cylindrical vessel of a size proportional to the power of engines, with a division plate going down to within 2in. or 3in. of bottom. In this compartment there may be one or two flat, thin perforated plates, and lying on top of these plates pieces of blanket, sweat rags, or fine gunny bags. The "feed" instead of being pumped direct from hot well to boiler, is discharged on top of the blankets, percolates through them, then through perforated plates, falls down to bottom, then has to rise 6in., where the feed suction takes hold of it, and from thence to boiler.

From this it will be seen that any grease, oil, or fatty matter is caught in the blankets and plates, and as they become saturated a hinged door on top is opened, the dirty cloths taken out and clean ones put in. This can be done in a couple of minutes. Passing the feed through charcoal in conjunction with this filter, and various other schemes, have been used with more or less success, and it is difficult to see why all feed water should not (comparatively speaking) be as pure as the water we drink.

The Greatest Possible Gain from Heating Feed Water.

The most efficient (or, at all events, the best known) feed-heaters in British practice are Weir's, Kirkaldy's, and Morrison's, and generally in all three the principle of heating is the same. Steam is usually taken from the low-pressure receiver; the feed water is forced by the feed pump into a chamber and is broken up into spray, or the "feed" is heated by steam coils suitably arranged; sometimes it

falls over a number of dash plates and drains down into the suction of another pump, which at slow speed discharges it into boiler.

The old practice of working feed pumps off air pump levers and discharging direct into boilers is almost obsolete, as it is much better in every way to discharge the quantity of water required at a slow than a quick speed, which would, of course, be the case if worked direct; besides, all air, oxygen, and deleterious gases have a better chance of being dissipated than with quick speed pumps.

The economy of feed-heating is a question about which much has been said and written, and many absurd results have been claimed and published.

Elaborate calculations have been made by Weir, Kirkaldy, and Morrison to account for the economy obtained, for when first introduced they did not look for or expect any saving, because the common sense view was they were robbing Peter to pay Paul; and without referring to their figures, I am of opinion that whatever economy is obtained is chiefly due to the fact that by raising the entering feed to 200° the natural circulation is only fractionally disturbed, and each particle of water does better work in proportion as the temperature of the entering feed approaches that of the steam; and when that point is reached, the circulation is undisturbed and consequently becomes more efficient, which would to a large extent account for the economy.

Feed-heating being of such importance, it is desirable to have a clear idea of what the saving would be for a given increase of temperature—that is, according to the laws which govern heat.

Example.

Take the normal temperature of water at	52° F.
We want to heat it up to, say	180° F.

Find total amount of saving.

The heat required to raise water from 0° to 212° is 1178 units. Then the number of units of heat required to convert one pound of water into steam at 212° when the normal

temperature is 52° will be $1178 - 52 = 1126°$. Now, if we heat this water by exhaust steam (which would otherwise be wasted), then the result would be all gain and we would have

$$1178 - 180 = 998 \text{ and } \frac{998}{1126} = \cdot 88 \text{ or } 12\% \text{ gain.}$$

Practically speaking, the economy of heating feed water in this way is seldom over 10 per cent.

In Kirkaldy's and Morrison's heaters, the steam, in passing through the coils, heats the feed to about 200° *when the coils are clean;* but in practical working it is found that when the coils become coated with oil and grease, their heating power becomes very much impaired, and on an average of, say, 40 days, although the temperature was 200° at the start, it is reduced to 150°, and even less at the finish, furnishing another good reason for the purification of the feed water.

Steam Engine and Boiler Efficiency— How to get at it.

When we want to find the comparative economy and efficiency of any steam engine, we have recourse to the "Indicator Diagram," which, when carefully taken and calculated, gives approximately the actual horse-power passing through the engines; but after taking diagrams, we find it necessary to go to the boiler to find what it has cost us in coal to develop this power. Then knowing the total amount of heat in the coal, we can very simply demonstrate the immense difference between theory and practice.

To illustrate: One pound of pure coal (carbon) contains 14500 units of heat (a British unit of heat is equal to raising one pound of water 1° F.), and when a pound of water is raised 1° we perform a certain amount of work. How much? An amount exactly equal to Joule's Equivalent—viz., 772lbs.—raised one foot high, and this shows that the

definition of a British unit of heat and "Joule's Equivalent" mean one and the same thing.

As there are 14500 units of heat in a pound of coal, and as each unit, if converted into work, would raise 772lbs. one foot high, we have $14500 \times 772 = 11194000$ feet, the height to which one pound would be lifted; or, to put it another way, if all the heat were utilised, the power developed would raise the pound weight over 2000 miles.

Then to find the actual horse-power per hour in a pound of coal, we have

$$\frac{11194000}{33000 \times 60} = 5{\cdot}65 \text{ H-P.}$$

Let us see how our Indicator Diagrams compare with the total heat in the coal. In the best designed Triple Expansion Engines, with a steam gauge pressure of 160lbs., it costs about $1\frac{3}{4}$lbs. of coal to develop one H-P.; and as it has been shown that if the total heat in a pound of pure coal could be converted into work it would develop 5·65 H-P., then we see that $5{\cdot}65 \times 1{\cdot}75 = 9{\cdot}88$, or, say, 10lbs. of coal is required to do what could be done by one, showing clearly that with all our knowledge we can only convert about 10 per cent. of the available heat into useful work. Summing this up, we might say that theoretically every pound of coal should give us 5·65 H-P.; practically we can only utilise $\frac{1}{10}$th of this, 90 per cent. of the total heat being lost.

Then comes the question, How is it lost, and what becomes of it? As will subsequently be shown, from 20 to 30 per cent. is lost in the boiler, but the greatest loss (so far as the heat is concerned) is in the engine itself, and this takes place at the point of exhaustion, either into a condenser or into the atmosphere.

As an example, and for the purpose of explanation, take a working pressure of 100 on steam gauge, the total heat in the steam would be $1185 + 32 = 1217$, and assume it worked the engine in the ordinary way, being expanded down so that it entered the condenser at atmospheric pressure, the total heat in it then would be $1146{\cdot}6 + 32 = 1178{\cdot}6$. The difference

in heat between the boiler and exhaust steam would be the difference between 1217 and 1178·6° = 39°, so that in working this engine we just manage to convert into work about 3¼ per cent. of the total heat in steam. This, of course, is an extreme case, but it will serve to show what an enormous quantity of heat is lost (less the amount represented by the temperature of the feed), and not only lost, but we have to construct expensive condensers, pumps, valves, levers, and other gear, to assist in getting rid of this heat; and so far as can be seen from the present state of our knowledge, we must keep on doing it.

From what has been said it will be clearly seen that "steam engine efficiency" is expressed by stating the amount of coal required to indicate one horse-power, and in our best practice it ranges from 1¾ to 1½ lbs. per indicated horse-power per hour.

"Boiler Efficiency."

The efficiency of a boiler is expressed in somewhat different terms to that of an engine. In both cases, however, it is the consumption of one pound of coal which governs the whole of the calculations; but before we can have a clear idea of this we must understand some of the properties of steam and water.

When water is boiled in the open air its temperature remains constant at 212° Fah., and any additional heat has no effect in raising its temperature. But it can be proved that a very large amount of heat has been put into the water which does *not* show on the thermometer, and this, for want of a better name, is called "Latent," or hidden heat. If we take one pound of water and convert it into steam at atmospheric pressure it has a sensible temperature of 212° and a latent heat of 966°—in other words, every pound of water in changing from water to steam absorbs as much heat as would have raised the temperature of the water 966° more than shows on the thermometer, provided it had not become latent.

This enormous amount of heat which has to be imparted to *each* pound of water represents $\frac{966 \times 772}{33000}$ =23 horse-power per pound, and the work done is expended in two ways:—1st, in overcoming the natural and mutual attraction of the water particles; and 2nd, in overcoming the resistance which the air or other medium presents to the formation of steam.

Expressing this in a practical form we might say that to tear the particles of water asunder and compel them to change their nature requires a force of 23 H-P., which must be developed in every pound of water converted into steam at atmospheric pressure. If we reverse the case and condense a pound *weight* of steam, the same reasoning holds good.

Many of the standard definitions of "latent heat" are vague, and to a mechanic scarcely satisfactory. One of the best (Maxwell's), by altering the words but not the sense, might be defined to be:—"The quantity of heat which must be put into every pound of water to convert it into steam *without changing the temperature.*"

From the above explanation we are now in a position to state that the evaporative efficiency of any boiler is expressed by the number of pounds of water which one pound of coal is capable of converting into steam under ordinary working conditions. Hence it will be apparent that if the total heat in the coal be divided by the latent heat in the steam the quotient will be the evaporative efficiency of the boiler expressed in lbs. of water converted into steam per pound of coal consumed.

Example.

Taking the heat units per pound as before, viz., 14500 units, we have $\frac{14500}{966}$ = 15lbs. of water evaporated per lb. of coal burnt—that is what we would get provided there were no loss. In ordinary working, however, we are a long

way short of this, and in the large majority of cases the evaporative efficiency is actually from 7 to 8lbs. How such losses occur will be subsequently explained.

It is very evident this "efficiency" depends very materially on the temperature at which the feed passes into the boiler, and as in practice this temperature has a considerable range, being often as low as 50 and as high as 200°, it becomes necessary in all evaporative tests to reduce them to a fixed standard, so that when comparing one boiler with another all the elements may be taken into consideration to enable us to judge correctly.

This is done by taking the actual measured evaporation at any pressure in lbs. of water per lb. of coal, and then finding what the evaporation would have been if the "feed" had been supplied and the steam given off at 212°, the difference being termed the "Standard Equivalent of Evaporation," and this is briefly expressed by saying that the steam is evaporated from and at 212°.

Example.

Assume we find by measuring the feed water and coal that a boiler evaporates a certain number of pounds of water per lb. of coal per hour, we want to know what the difference would be in the evaporation if the temperature of the feed had been 212° and the steam also generated at 212°. To find this we have:—

$$\frac{\text{Total heat in steam} - \text{temperature of feed}}{966} = \text{factor of evaporation.}$$

Example.

A boiler with 200lbs. steam gauge pressure evaporates 8lbs. of water for every pound of coal burnt per hour; feed is supplied at 150°. Find the evaporation from and at 212°.

Total heat in steam at 200lbs. per sq. in. is $1200+32=1232°$.

therefore $\frac{1232 - 150}{966} = \frac{1082}{966} = 1{\cdot}12$, the "equivalent of evaporation."

And $8 + 1 \cdot 12 = 9 \cdot 12$lbs. of water evaporated from and at 212°.

Then in the same example, to find how much of the total heat in the coal is utilised in converting the water into steam when the steam gauge shows 200lbs. and the measured evaporation is 8lbs. of water per lb. of coal per hour, we have $8 \times 966 = 7728$, the heat units utilised—therefore $\frac{7728}{14500}$ $= \cdot 53$ per cent., being 47 per cent. less than what the total heat units in the coal would give provided there had been no loss.

In the foregoing explanation we have given the coal as practically pure, giving it credit for 14500 heat units per pound, but even the best coal contains considerably less heat than this, and the two following examples will give approximately a better idea of what we get in actual working, the first showing the best and the second the worst results:—

1st. Assume a boiler of the best design and that good coal is used, every pound of which can, theoretically, evaporate 14lbs. of water from and at 212°; also, that the actual measured evaporation from and at 212° is 11lbs. of water per lb. of coal per hour. We want to know the percentage of efficiency, and also the percentage of loss.

$$\text{Here } \frac{11 \times 966}{14 \times 966} = \frac{10626}{13524} = \cdot 78 \text{ per cent. efficiency.}$$

or, what is the same thing, $\frac{11}{14} = \cdot 78\%$.

The loss is $100 - 78 = 22$ per cent.

2nd. Take the same boiler, using very bad coal, whose theoretical evaporation is only 9lbs. from and at 212°. Then if the actual measured evaporation from and at 212° be 5lbs. of water per lb. of coal per hour, we want to find the respective percentages of efficiency and loss.

In this case we have:—

$$\frac{5 \times 966}{9 \times 966} = \frac{8694}{4830} = \cdot 55 \text{ per cent. efficiency.}$$

and $100 - 55 = 45$ percentage of loss, and this is about

the average of what we may expect to get from good or bad coal.

Referring to evaporative tests generally, it should be remembered that from a practical point of view many of the published results must be received with a considerable degree of caution, because, unless the steam is actually dry, a certain quantity of water may be carried over with it, and then the boiler is credited with an evaporative efficiency to which it is not entitled. This is more likely to take place when the boiler is small for its work or badly proportioned, and especially if the area of water line is too contracted and the capacity of steam space limited.

To get at anything approaching the true evaporation requires considerable knowledge and experience, and, in any case, all such tests cannot be carried out without extreme care and expense.

Specific Heat as Applied to Iron, Steel, and Water in Boilers.

As before stated, a British unit of heat is the amount required to raise one pound of water 1° Fah.

This is the standard by which we compare the amount of heat required to raise the temperature of any other substance 1°, and whatever the difference may be is termed the "Specific Heat" of such substance compared with an equal *weight* of water. The "specific heat" of the metals compared with water is as follows:—

Water at 32° Fah. is	1·0000
Cast iron at 32° Fah. is	·1298
Boiler steel and wrought iron is	·1152
Copper, gunmetal, and brass is	·0952

Water is selected as the standard because it has a much higher specific heat than anything else known (except hydrogen), whereas the specific heat of all metals is very low.

The following example will give a practical grip of the principle of specific heat:—

Assume we have to find the difference in the amount of

heat required to raise the same weight of iron and water to the same temperature, say, 212°.

Weight of iron is	..	112lbs.
Weight of water is	..	112lbs.
The specific heat of water is	..	1
The specific heat of iron is	..	·1152

Therefore we have 112lbs. of iron $\times$ 212 $\times$ ·1152 = 2735 heat units required for the iron, and 112lbs. of water $\times$ 212 = 23744 heat units required for the water, showing that in heating the same weight of water and iron to the same temperature, it takes about 9 times more heat for water than for iron.

$$\frac{23744}{2735} = 9 \text{ times nearly.}$$

As a rough and ready rule, it may be said that to raise the temperature of a given weight of iron, steel, copper, gunmetal and brass requires from 9 to 10 times *less* heat than to raise the temperature of the same weight of water.

Specific Heat of Coal.

The specific heat of average coal is ·2412, usually taken as being $\frac{1}{4}$th that of water, consequently, taking previous example—viz., raising the temperature of equal weights of iron and water to 212°—and applying the same to coal and water, we would have:—

112lbs. coal $\times$ 212 $\times$ ·25 = 5936 units of heat required,

and 112lbs. water $\times$ 212 = 23744 units of heat required,

showing it takes 4 times more heat to raise the temperature of the water than the coal, all other things being equal.

We burn coal in a furnace for the purpose of transmitting all its available heat energy to the water in the boiler; and as the natural consequence of confining boiling water is to generate steam, we do our best (which is not much) to convert all this heat energy into mechanical work, and it is important to remember and realise that the higher we get the initial temperature and the lower we get the temperature

of the exhaust, the greater will be the efficiency of both engine and boiler—or, what is the same thing, as the initial steam pressure is increased and the exhaustive pressure decreased, the greater will be the efficiency and economy of our machinery.

Total Heat in Steam—To find it without the Tables.

When we increase the steam pressure does the total and latent heat remain constant? No; it does not. The total heat is increased and the latent heat becomes less.

In the old days, before the experiments of Regnault and others, it was assumed that the latent heat was a constant quantity. Now we know better, and as the pressure rises the latent heat is less and the total heat more, but not much. How much? The total heat in steam at 212° is 1146°, and from experiment it has been found that for every degree of temperature above 212° the total heat in steam increases ·305°. Therefore, if we multiply the number of degrees over 212° by ·305, and add the result to 1146, we get the total heat in steam at any pressure.

Example.

Take a boiler with a working pressure of 180lbs. on steam gauge. We want to find the total heat in the steam?

Steam at 180lbs. has a temperature of 379° Fah.
Therefore 379 — 212 = 165° over 212°.
And 165 × ·305 = 51° more heat in it.
Then 1146 + 51 = 1197°, the total heat in [steam at 180lbs. pressure.

Take another example, where the steam pressure is only 20lbs.

Steam at 20lbs. pressure has a temperature of 259°.
Therefore 259 — 212 = 47° over 212.
And 47 × ·305 = 14° more heat in it.
Then 1146 + 14 = 1160°, the total heat in [steam at 20lbs. pressure.

Temperature of the Fire in Furnaces.

It is by no means easy to get at the temperature with any certainty, as it may, and does, vary considerably according to the quantity and quality of coal, the amount of air supplied, the force of the draught, etc. The instruments used (pyrometers) for determining such temperatures also present a difficulty, as the results obtained are by no means reliable. The following may, however, give a general idea of what is received as being approximately correct.

Assume that our furnaces are so arranged that in burning good coal we get perfect combustion, then under such conditions we would get the highest possible temperature. The result of burning one pound of coal perfectly would be 2·93lbs. of carbonic acid, ·45lbs. of steam, and 8·49lbs. of nitrogen, equal to 11·87lbs. altogether. The specific heat of coal is approximately $\frac{1}{4}$th that of water, therefore $11{\cdot}87 \times {\cdot}25 = 2{\cdot}9$ units of heat required to raise the temperature of the furnace 1°; and as the total heat of combustion is, say, 14500°, we have $\frac{14500}{2{\cdot}9} = 5000°$, the theoretical maximum temperature of furnace, allowing the air to have been supplied at a normal temperature of 60°.

The published records relating to furnace temperatures are limited, and vary considerably. Even D. K. Clark, in his excellent work on "Steam Engines and Boilers," treats this question empirically—that is, he assumes most of the data necessary to work out the calculations, and embodies his results in a tabulated form, which is probably not far from being correct. Clark's table shows that although when burning from 5 to 10lbs. of coal per square foot of grate bar the temperature is comparatively low, from 1400° to 1500°, yet with a high rate of combustion, from 40 to 120lbs. per square foot, the temperature does not increase materially, and may be set down at an average of 2000° Fah.

There is a considerable difference in the temperature of marine furnaces, and those encased in brick, whose internal

surfaces are usually carefully coated with some good non-conducting material, the object of which is to conserve and confine the radiant heat from the coal, and they are arranged in a way that such radiant heat shall, as it were, be returned to the fire by counter radiation, as it cannot get away. It is in this manner and by such means that reverberatory and other furnaces are kept and maintained at such high temperatures. In return tubular boilers, with internal furnaces, the conditions are very different. The radiant heat in such cases is not conserved, confined, or given back to any great extent, but is quickly absorbed, passing rapidly through the surface plates (whose temperature on both sides is assumed to be the same, or nearly the same, as the water in the boiler), making it impossible to keep up, or even approach, the theoretical temperature of combustion. Taking, as shown, 5000° as the maximum of perfect combustion and 2000° as the average of practical working, we get $\frac{2000}{5000} = \cdot 40$ per cent. of the theoretical temperature of perfect combustion in internally fired furnaces with natural draught.

How the Force of the Draught is Calculated.

The draught power of a chimney or funnel depends on the height and is independent of the internal area, and its efficiency is greatest when the volume of the escaping gases is double that of the external air—that is, when the hot gases inside are about one-half the *weight* of the air outside. To obtain such conditions, the temperature of the outside air would be 60° Fah., and that of the gases in tunnel 552° Fah.; then if the height of the funnel or chimney (measuring from fire bars) be 100 feet, such gases have a velocity due to a *head* of $\frac{100}{2} = 50$ feet.

To show how the force or power of this or any other

draught, either natural or forced, may be calculated, consider a column of water 100 feet high; the pressure on bottom of column would be $\frac{100}{2 \cdot 3} = 43 \cdot 5$lbs. per square inch, because a column of water 2·3 feet high and a square inch in section weighs one pound, and as air is 820 times lighter than water, the pressure on the bottom of an air column 100 feet high must of necessity be 820 times less than on the water column, and, as in the above example, the weight of the escaping gases is only one-half that of the external air, our air pressure or force in funnel must be $820 \times 2 = 1640$ times less than water. Therefore we have 100 feet $\times$ 12in. $= 1200$in. high, and $\frac{1200}{1640} = \cdot 75$in. or $\frac{3}{4}$in. of water, the force or power of the draught—that is to say, the force of the draught in this chimney is equal to supporting a column of water $\frac{3}{4}$in. high.

Heat Lost Through Draught.

The previous example shows clearly that we can only produce the required draught by the expenditure and loss of a considerable amount of heat. To obtain a good draught means that the temperature of the escaping gases in chimney must not be less than 500° Fah., and if we take the temperature of the furnace at 2000, then a rough and ready rule easily remembered would be to divide the furnace by the funnel temperature, the result being what we lose—thus $\frac{2000}{500} = \frac{1}{4} = 25$ per cent.

This rule, however, is scarcely sufficient, and the following is the usual method of calculating the loss:—

Multiply the weight of air and coal by the difference of temperature between inside and outside of funnel, and also by the specific heat of air. This, divided by the total heat in one pound of coal, represents the loss.

Example.

Assume it takes 24lbs. weight of air to burn one pound of coal; the result is 24 + 1 = 25lbs. of gases. And if the difference in temperature between inside and outside of chimney were 550° Fah., then, taking the specific heat of air at $\frac{1}{4}$th that of water and the heat units in one pound of coal at 14500, we have $\frac{(24 + 1) \times 550 \times \cdot 25}{14500} = 23 \cdot 7$ per cent. of the heat carried off and lost in producing the draught.

The increase of draught due to extra height may be approximately stated to be equal to supporting $\frac{1}{16}$ in. of water for every 10 feet of height—that is, if a chimney 60 feet high had a draught equal to $\frac{7}{16}$ in. of water, then by making it 70 feet high the force of the draught would support $\frac{8}{16}$ in. (or $\frac{1}{2}$in.) of water.

Modern Forced Draught.

When coal is burnt under ordinary conditions—that is, with natural draught, the amount consumed is about 16lbs. per square foot of fire grate; and if we want to get more power out of a boiler, the force of the draught is increased by mechanical means. This is usually done by the introduction of fans, which force the air at the required pressure, either under or over, and sometimes both under and over, the fire. When the force of the draught and the heating surfaces are arranged proportionately to burn two, three, or four times the quantity that can be consumed with natural draught, then, according to the theory of the principle, we should, and would, get two, three, and four times as much power from every square foot of grate bar—that is to say, if four times 16 or 64lbs. of coal is consumed per square foot of grate, then one boiler would do the work and develop the same power as four boilers working under ordinary conditions with natural draught. But, like many other things pertaining to engineering, there is a wide difference between theory and practice, as for various rea-

sons it is found that the intensity of this draught is limited, and anything in excess of a force capable of supporting a column of water 3in. in height gives no end of trouble—in fact, this force in the *ash-pits* seldom exceeds $1\frac{1}{2}$in. of water, as the height of column at the fan is generally reduced (through friction, etc.) from 30 to 40 per cent. before it gets to the fire bars.

The causes which restrict and limit the force of the draught are various, but intense local heating, inrush of cold air, cleaning and banking fires, sweeping tubes, extreme variations in temperature, defective design, bad circulation, etc., have been assigned as reasons for over-heated furnaces, springing of tubes, burning of tube ends, bulged tube plates, sprung landings, leaky rivets—in fact, actual experience has demonstrated that when forced draught is used it must be like boiler steel, very mild indeed.

There is still a considerable difference of opinion amongst engineers as to its efficiency or otherwise. Without discussing it in detail, there is one thing which materially affects those in charge of it, and it is this—that builders and owners in many cases scarcely realise the absolute necessity of special provision being made for an ample supply of really pure water in making up the loss. Evaporators, as a rule, are in nine cases out of ten far too small for their work, and in those fitted with "coils," to prevent priming, the engineers are obliged to keep the water line so low as to leave at least one-half of the heating surface totally uncovered, and even then a large majority of them prime badly. What appears to be one (if not the principal) fault is that in all such machinery the area of water level is contracted; the steam generated cannot leave the surface freely, but lifts and carries a certain percentage of salt water direct into the condenser, and in such cases it is more or less a trap, which, unless well watched, may with an intense forced draught do an enormous amount of damage. Another thing which appears to have received but scant consideration is that the *area* of water level required for such a low pressure

requires to be very much greater (as previously shown) than for a high pressure; and if the sectional area in our evaporators were increased 50 per cent., we would probably have less trouble; and if in conjunction with this the purification of the "feed" received the attention it deserves, we should be able to work this forced draught with a great deal more confidence and satisfaction.

It should always be remembered that if we burn 32 instead of 16lbs. on every square foot of grate we tax the absorbing power of the water in the proportion of 2 to 1—that is, the water has to absorb double the amount of heat *in the same time,* and personal experience has proved that unless the internal surfaces are kept quite clean, and especially furnace crowns, the plates will be overheated.

This has been practically demonstrated in boilers carrying 160lbs. of steam, where the force of the draught in *ash-pits* was not more than 1½in. of water, and a lime scale on furnaces no thicker than a sixpence.

But it may be asked—How about a locomotive where the amount of coal burnt is often as much as 80 to 100lbs. per square foot of grate, and where the force of the draught is more than double that used in marine practice? This is a very pertinent question. But it must be remembered that in a locomotive the fire-box is generally of copper, which is a much better conductor of heat; the surfaces in fire-box are all flat, and stayed every 4½in.; that the water is generally pure and clean; that where impure water is used the result is the more or less bulging of plates between the stays, which under exceptional circumstances (if the water is very bad) may split the plate. But in marine furnaces, and especially those of large diameter, it is very different, so far as safety is concerned, because the "feed" is never pure and clean, and as the strength of such furnaces depends almost entirely on their being and remaining—comparatively speaking—true circles, any overheating or change of shape is of greater importance and more dangerous than in furnaces constructed as in locomotives.

WATER TUBE BOILERS.

In Marine practice the size and *weight* of the ordinary tubular boiler have increased to such an extent that the question of a more suitable design has for some years past been attracting considerable attention. There can be no doubt that large boilers of the cylindrical type will not safely carry much higher pressures than they are now subjected to. Some design other than the ordinary cylindrical boiler must be adopted when the pressure exceeds 200 lbs. per square inch as the thickness of the shell plates and furnaces required would, with the larger diameter boilers necessary in order to save space, make them almost impracticable. When we consider that in some of the most recent specimens of what are termed "Atlantic Greyhounds" the boilers are 17 ft. 3 in. diameter by 22 ft. long and that the weight of each boiler is about 110 tons (without water) it is only natural to conclude that we must be getting pretty near the limits of diameter and thickness; hence it is that some of the most capable engineers are of opinion that a change in design is not only desirable but necessary, and that the modern water tube boiler is the most likely to comply with the requirements.

The water tube boiler is no new idea; its history dates back to early in last century. Scores of patents have been taken out, but with a few notable exceptions the results have been disappointing, few, if any, coming up to expectations. In 1857 they were tried in the Mercantile Marine but were not successful. As a result of increased experience of higher pressures renewed attemps were made about 1870-74 to obtain such boilers, but again without success; and for some years after this the attempt in Great Britain was practically abandoned.

In France the trials were continued, with the result that in 1880 a despatch vessel was fitted with Belleville water tube boilers, and was employed in actual sea service to a considerable extent. In 1882 the French cruiser *Milan* was fitted with similar boilers. Experience with these vessels

seems to have been satisfactory and the Belleville boiler has since that time been extensively adopted in the French Navy.

Soon after the trials in the French Navy the Messageries Maritimes Co. fitted one of their vessels with these boilers, followed in 1886 by a mail vessel of 2,400 I.H.P., and, as a result of this experience, boilers of the same type have since been fitted to their largest and fastest mail steamers.

During this time the British Admiralty had been carefully watching the progress of water tube boilers, and as far back as 1877 the first Boiler Committee recommended their adoption, but nothing came of it. However, when favourable information was received regarding the successful working of Belleville boilers in the Mail steamers of the Messageries Maritimes Co. running between France and Australia, an engineer officer was sent to report on their actual working on a voyage to Australia and back. His reports were favourable, and were subsequently confirmed by the observations and reports of other Admiralty officers. The Admiralty thereupon decided to fit the large cruisers *Powerful* and *Terrible,* of 25,000 I.H.P. each, with Belleville boilers, and in 1897 they had ten cruisers either built or building having an aggregate of 156,000 indicated horse-power fitted with these boilers.

The "principle" of all (or nearly all) water tube boilers is practically the same, the only difference being in the design and arrangement of the mechanical details.

A large class of water tube boilers have an upper chamber into which the steam and water are delivered by a series of tubes surrounding the fire and connected to a bottom chamber or chambers. These chambers are also connected with large downcast tubes which serve to keep up the circulation through the heating tubes.

Another type consists of horizontal or nearly horizontal tubes placed above the fire connected to suitable boxes or headers at each end. On the top of these boxes or set of headers is the steam and water drum, from which a suitable

connection is made to the box or header at the other end for proper circulation.

This class of boiler has the great advantage that the tubes can be easily examined, cleaned or removed.

When the heated water in the tubes begins to circulate, being lighter, it moves upwards into the front headers, to which are connected vertical tubes whose top ends are jointed to a horizontal steam chest, which is usually half full of water. The steam space of the boiler is represented by the capacity of the top half of this receiver. The water in the lower half flows into a corresponding number of tubes at the back end, then into the heating tubes again, when the same operation is repeated. Sometimes, instead of vertical tubes at front and back, narrow water spaces are fitted and stayed in a similar manner to the back water spaces in a single ended tube boiler, but the "principle" so far as circulation and generation of steam are concerned is the same. The heated water in the tubes over the fire discharges into the front space, then into the receiver, out of receiver into back water space, thence into the tubes over the fire again, thus completing the cycle.

In the past a large majority of water tube boilers were expensive and troublesome, so much so that in many cases they were discarded.

Numerous objections were taken to them, such as complication of parts, enormous number of joints which were awkward to make and difficult to keep tight, imperfect circulation, corrosion, disproportion and inefficiency of heating surface, impure water causing deposit, limited provision for cleaning resulting in bent or split tubes, cramped area of water surface causing priming, large space required and great weight of brickwork, etc.

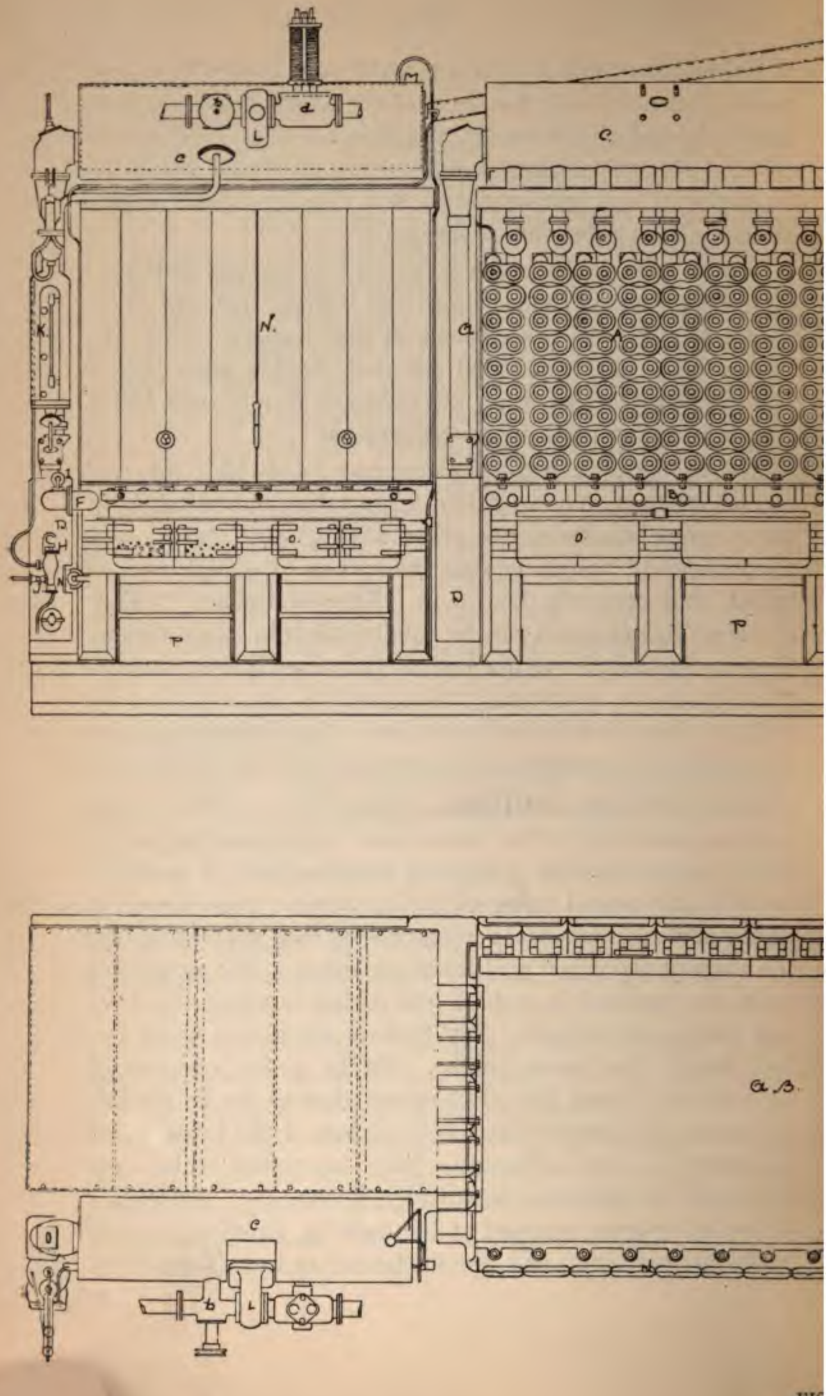
The advantages claimed are principally increased safety, less first cost, less weight, a practically unlimited pressure, rapid and complete circulation with a quantity of water whose weight is only fractional compared with that in an ordinary boiler, facility and economy of repairs, etc.

There is no doubt that of late water tube boilers have been vastly improved and that most of the above objections have been successfully overcome, and this has only been accomplished through expensive experience, increased excellence in material and workmanship, and also by drastic alterations and improvements in design.

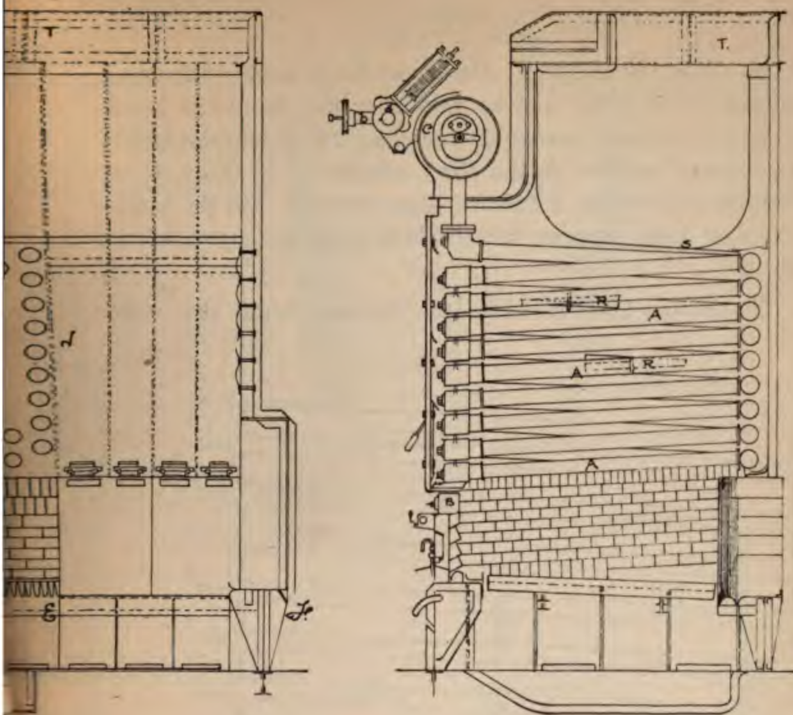
Water tube boilers cannot be said to be any better as steam generators than cylindrical boilers, nor can it be said that they are less efficient in that respect. They are, however, 20 per cent. to 40 per cent. lighter, according to the pressure carried, than the ordinary return tube boiler, which in many cases is a consideration.

The modern water tube boiler may be divided into two classes, the one having fairly large tubes and suitable for ocean-going steamers, the other having small tubes, only used in small vessels having short runs at a high rate of speed, and generally known as "Express Boilers." Each of these classes again can be subdivided into two divisions, the one having the upper end of the tubes under water or what is known as drowned tubes, and the other having the ends of the tubes above water level, the circulation being maintained by priming and discharging into the steam space.

BELLEVILLE BOILER. (Fig. 39.) This boiler consists essentially of a steam and water chamber and a lower water chamber connected together with a series of straight zig-zagged tubes of comparatively large diameter. There is also an external return tube on each side connecting the ends of the upper and lower chambers. The generating tubes are enclosed in a sheet iron casing in which the flame and gases are confined. The firebars are placed about two feet below the lowest tubes. Baffle plates are secured amongst the tubes, placed to secure that as far as possible the gases will traverse the whole surface of the tubes before escaping to the chimney. The generating tubes are arranged in sections, technically known as "elements." These tubes are connected at the ends by junction boxes of malleable cast iron, and are zig-zagged so as to form a flat



FIG



A. Steam Generator Element.
B. Feed Pipes. (Collectors).
C. Collectors & Purifiers of Steam & Feed Water.
D. Mud or Dirt Box.
E. Return Pipes from Purifier to Mud Box
F. Connection from Mud Box "D" to Feed "B"
G. Graduated Feed Cock.
H. Blow Off Cock
I. Salinometer Cock
J. Injection Cock. (direct)
K. Feed Regulator.
L. Steam Pipe Connection to Drum, Safety & Throttle V
M. Test Cocks.
N. Sheet Iron Casing & Brick Lining to Furnaces
O. Furnaces
P. Ash Pit Doors
Q. Steam or Air Jets in Furnaces.
R. Baffle Plates to Regulate Passage of Gases B
S. Baffle Plate on Top of Elements.
T. Funnel Base.

ILER.

spiral (Figs. 40 and 41). The front boxes have a hand hole opposite each tube, and by removing the doors the inside of all the tubes is accessible for cleaning and examination. An element as fitted in the Navy consists of 20 tubes, $4\frac{1}{2}$ in. diameter and about 7 ft. 6in. long, screwed into the malleable cast iron junction boxes which form the turns of the spiral.

The connection between the element and the water

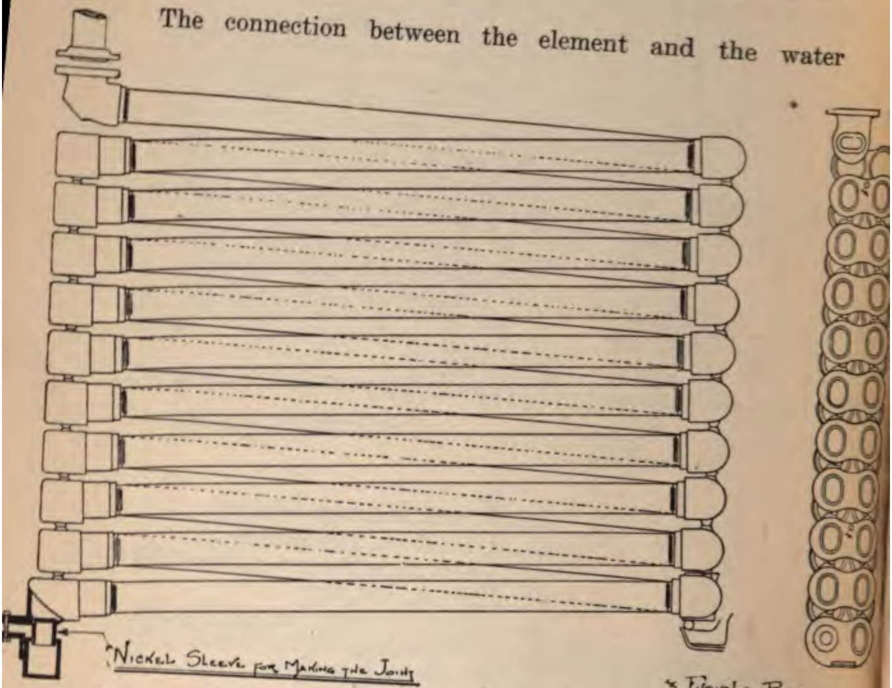


FIG. 40.—BELLEVILLE BOILER ELEMENT.

chamber is made by means of a coned nickel sleeve secured by one bolt. The top of each element is joined to the upper chamber and has a short internal pipe, and inside the chamber a series of dash plates are fitted for separating the steam from the water. The feed water is admitted through a small check valve on the steam drum, where it is discharged through a small orifice at a much higher pressure than that inside the boiler, and mixing with the other

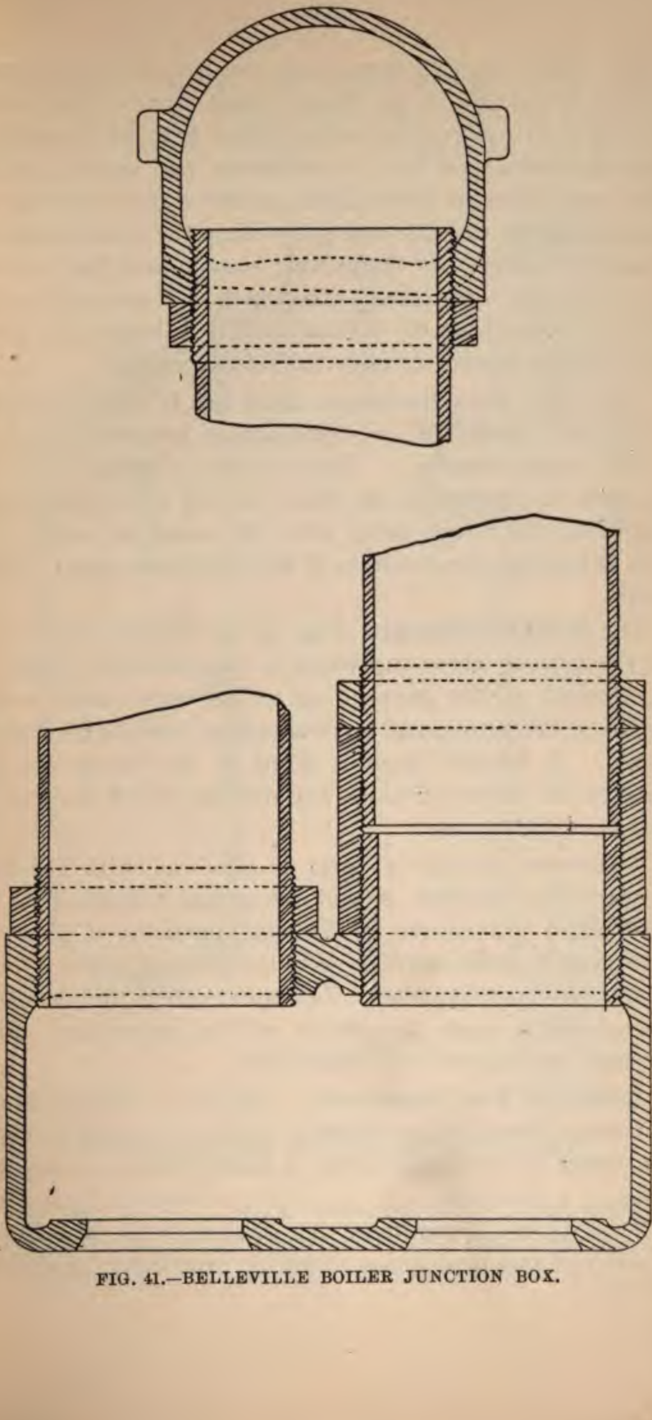


FIG. 41.—BELLEVILLE BOILER JUNCTION BOX.

water in the drum it flows along the bottom to the return tubes. The water in the steam drum, owing to the admixture of feed water, is of course colder than the steam that is being discharged from the elements, and special precautions are taken to prevent this colder water entering the elements from the top and preventing the continuous discharge of steam. It is for this reason that the internal pipes already mentioned, and which are carried up for about 8 inches from the bottom, are fitted, leaving the lower part of the chamber free for the return water.

The water from the return tubes has to pass through a non-return valve to a sediment chamber before entering the lower water chamber. This non-return valve is for the purpose of preventing the water leaving the elements and ascending the return tubes when the vessel is rolling, and also to regulate the direction of the circulation when raising steam.

The Sediment Chamber (Fig. 42) has a fairly large space at the bottom where the water is comparatively quiet, so that nearly all the grease, lime, or any other solid matter settles at the bottom and so does not get into the generating tubes. A blow-off cock is fitted to the bottom of this chamber by means of which any deposits which accumulate there are blown out.

A certain quantity of lime is admitted with the feed water. The intention is that the grease contained in the feed water will adhere to the small particles of lime and thus acquire sufficient density to quickly settle as a deposit in the Sediment Chamber. To a great extent this does take place, and a large proportion of the grease and other deposits are trapped and blown out.

Automatic Feed Apparatus. (Fig. 43.) This is a most important fitting on a Belleville boiler, by means of which the water in the gauge glass is maintained at a constant level.

Without this apparatus or its equivalent the working of a boiler of this description would be practically impossible.

As previously mentioned, the feed pumps are proportioned so as to maintain a much higher pressure in the discharge pipes than exists in the boiler. For a boiler pressure of

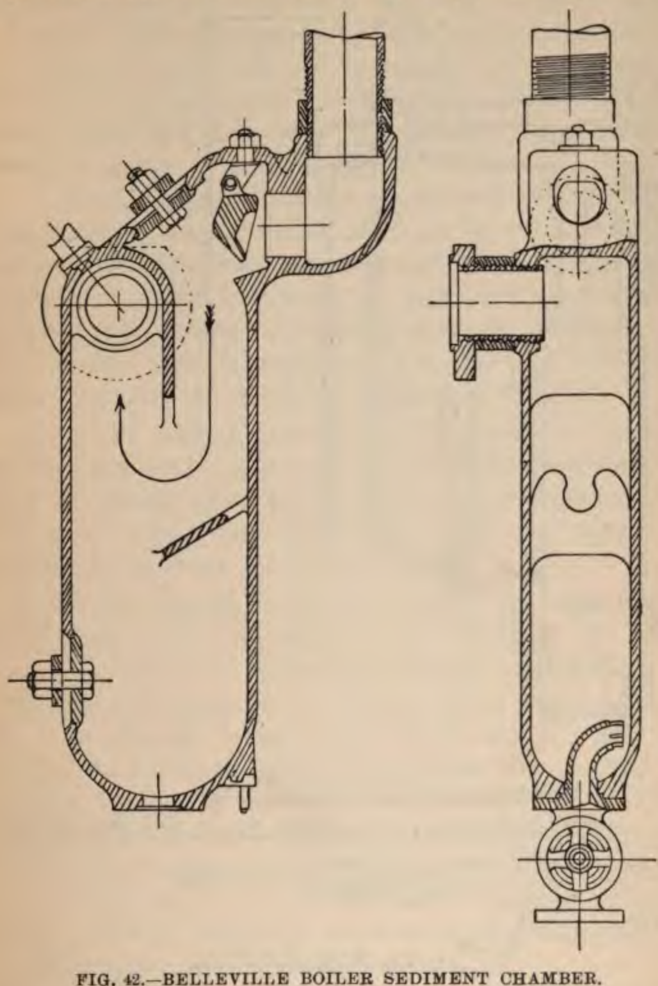


FIG. 42.—BELLEVILLE BOILER SEDIMENT CHAMBER.

250 lbs., the pressure at the pump would be about 600 lbs.

The automatic feed regulator will be easily understood by referring to the drawing. It consists of a chamber

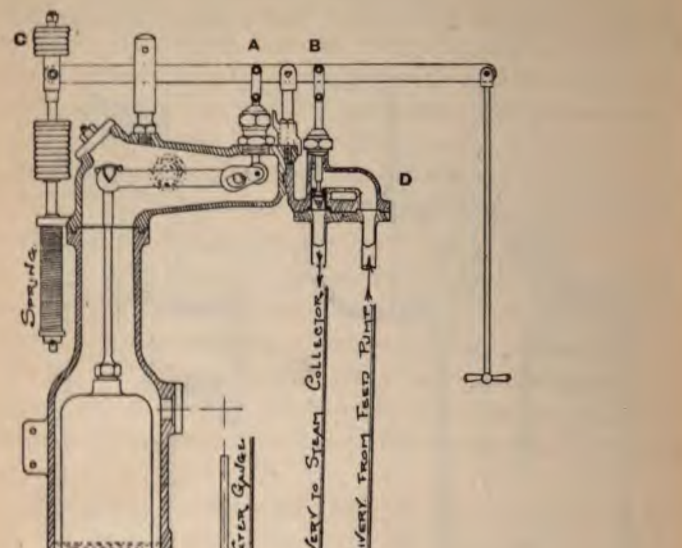


FIG. 43.—BELLEVILLE BOILER AUTOMATIC FEED.

which is connected at the top and bottom with one element of the boiler, and to this chamber one of the gauge glasses of the boiler is attached. The chamber contains a float which is moved up or down by the water contained in the chamber. This float has to be of a substantial character to withstand the external pressure, and is balanced by outside weights (C) and spring so as to enable it to float freely when half immersed. The delivery from the feed pump is carried to a valve which is connected to the float by a system of levers and the weights and spring already mentioned, in such a manner that, as the water level rises and lifts the float, the external weights and spring acting on the end of the lever close the valve and the admission of feed water to the boiler stops until the water level in the chamber again falls far enough for the float to descend and again raise the weights, thus opening the valve.

It will be noticed that there is no connection between the ingoing feed water and the regulating chamber, the feed valve being bolted to it only as a matter of convenience. There is a direct connection between the feed regulating valve and the check valve on the boiler.

Great care has to be taken in fitting the parts of this gear, knife edges are fitted to all bearings, and every precaution taken to reduce the friction to a minimum.

As already explained, the forces tending to open or shut the feed valve are brought about by the action of the float due to the rising or falling of the water level beyond the water-borne position, increased by the internal lever. There are, however, several other forces acting on the valve, viz., the boiler pressure on the spindle A, the feed water pressure on the valve at D, and the feed water pressure above the valve, which as previously stated is much greater than the boiler pressure.

We will now consider the action of the whole of these forces on the valve when the float is just water-borne and the valve shut. Under these conditions the pressure between the valve and the boiler will be practically the same as the

boiler pressure. The bore of the feed valve and the spindle A are made the same diameter, and, having the same leverage and being subject to the same pressure, these forces will neutralise each other. The pressure of the feed water above the valve has no effect because the spindle B is made the same diameter as the bore of the valve; therefore the pressure upwards and downwards is equal, so that no matter what the pressure may be between the pump and the regulating valve it has no effect upon the valve.

It will thus be seen that if the gear is properly designed and the valve is just shut everything is in a state of equilibrium, and the only force tending to open the valve will be that due to a deficiency of buoyancy of the float. When, however, the valve is open the feed pressure between the valve and the boiler becomes greater than the boiler pressure and tends to close the valve; the water level has therefore to fall a little to overcome this to keep the valve open.

Water Level. In the ordinary cylindrical boiler under working conditions the water in the gauge glass shows approximately the water level inside the boiler, but in the Belleville boiler this is not so, the water level in the glass depending entirely on the rate of evaporation. The gauge connections—or, what is the same thing, the feed regulator chamber connections—are attached to the bottom tube junction box and the seventh from the bottom. The gauge will therefore indicate what takes place in that portion of the element, the total height of the element being ten pairs. This position has been found by experiment to give the correct *amount* of water inside the boiler when the water in the gauge glass is showing at the working height.

It must be remembered that when the boiler is at work the water in the tubes is not solid to what may be termed the water level, but the whole of the tubes are full of a mixture of water and steam, the steam gradually increasing in volume as it ascends the element and becoming greatest in the upper tube. The water in the gauge glass really represents the difference of pressure due to the mixture of

steam and water at the two points of attachment, and also the excess pressure at the lower tube necessary to cause the water and steam to flow from one point to another through the tubes. The water showing in the glass therefore does not show the water level inside the boiler except, of course, when the boiler is at rest, but at any given rate of evaporation it is simply a measure of the quantity of water inside the boiler.

It has been proved by experiment that when the water level is about half way between the two connections there is always sufficient water in the mixture to prevent overheating of the tubes.

As the rate of evaporation is increased it is evident that the velocity of the water and steam in the tubes likewise increases, and, as a consequence, there must be an increase of pressure at the lower tube to insure this flow. This increase of pressure in the lower tube would of course cause the water level in the chamber to rise although the amount of water in the boiler is not increased.

As the feed regulator is designed to shut off the feed at a constant level in the chamber, it is evident that the amount of water in the boiler must vary with the rate of evaporation. The question naturally arises, what is the actual amount of water left in the boiler at different rates of evaporation? To settle this question a boiler was experimented with, and the following results obtained, the water in the gauge glass in each case indicating the same level:—The boiler with cold water level at working height held 3,000 lbs. water; when burning 12 lbs. coal per sq. ft. of grate the amount of water had been reduced to 2,400 lbs.; at 20 lbs. per sq. ft. it was 2,100 lbs.; and at 30 lbs. it was reduced to 1,600 lbs. By the above figures it will be seen that the amount of water in the boiler is almost exactly in proportion to the rate of evaporation.

A study of the above will at once show the reason why it is that the feed pumps attached to these boilers either go

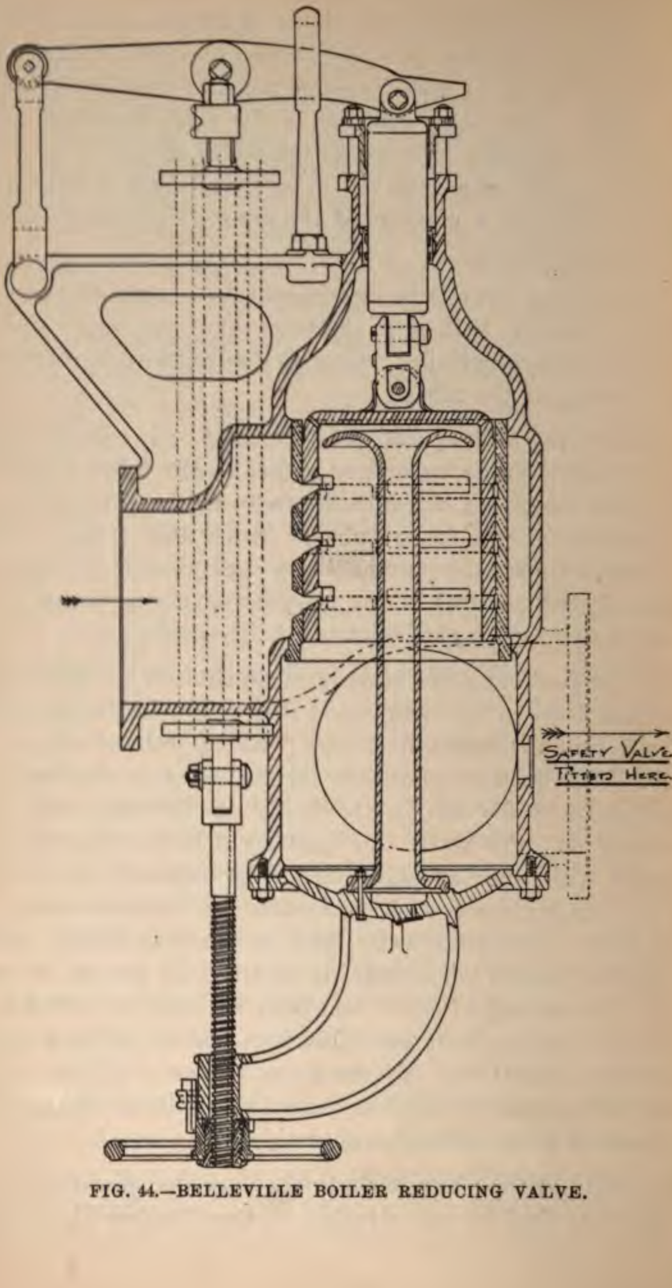


FIG. 44.—BELLEVILLE BOILER REDUCING VALVE.

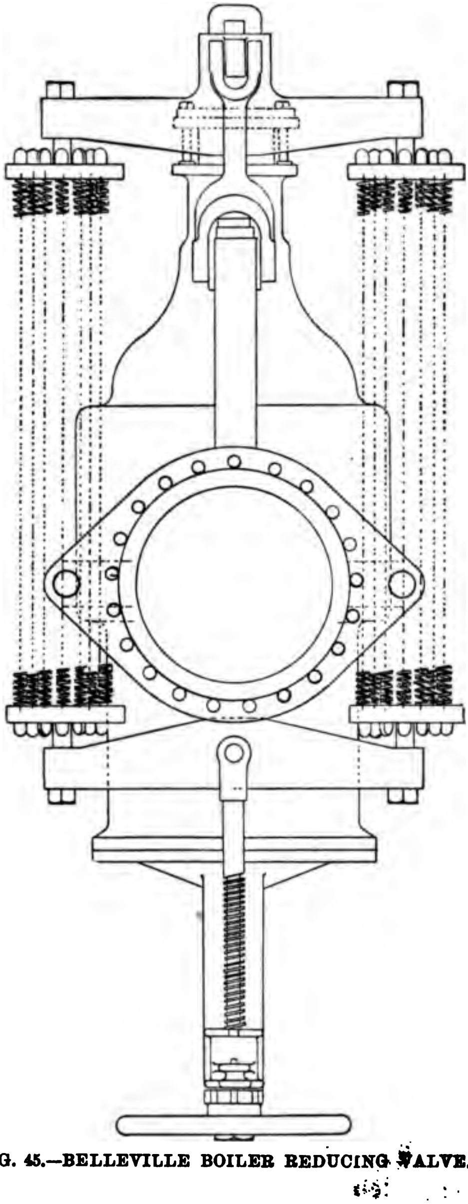


FIG. 45.—BELLEVILLE BOILER REDUCING VALVE.

slow or stop when speed is being increased, and work rapidly when slowing down.

Reducing Valve. (Figs. 44 and 45.) The working pressure carried on the Belleville boiler is always arranged to be about 50 lbs. per sq. in. above the pressure required at the engines. The steam is supplied to the engines through a reducing valve which wire draws the steam; this has the effect of drying up any moisture contained in the steam before reaching the engines. Owing to the small quantity of water contained in the boiler and the very limited steam space, fluctuations of pressure are much more liable to occur than with a cylindrical boiler; the reducing valve therefore becomes a necessity so as to maintain a steady pressure at the engine stop valve.

The reducing valve is simply a bell-shaped piston valve worked by a spring-loaded plunger. The piston has a number of small ports, and there are corresponding ports in the valve liner. The load on the plunger is so arranged that when the pressure in the steam pipe reaches the required amount it overcomes the load due to the springs, and closes the ports. These valves are very efficient, and the pressure at the engines is almost constant.

Between the boiler and the reducing valve a large automatic separator is always fitted, and discharges the water separated from the steam into the condenser.

All the work on a Belleville boiler has to be of a very high character, which makes it costly to manufacture, and even with all the care taken it loses so much water through numerous small leaks that its efficiency is seriously impaired.

BABCOCK AND WILCOX BOILER. (Figs. 46, 47, and 48.) This boiler consists of an arrangement of inclined tubes, sinuous chambers or "headers," to which the tubes are attached, a horizontal steam and water drum, and a mud drum. The furnace is, of course, below the inclined tubes, and is usually the full width of the boiler.

The tubes are placed at an angle of 15 degrees from the horizontal to ensure a continuous circulation in one direction, and are divided into vertical sections or elements. Each section is made up of a series of straight tubes expanded into the sinuous steel headers. These headers are of square section, and made of solid steel welded up and neatly finished.

The steam and water drum, which is of ample size, extends across the front of the boiler, and is connected to the upper ends of the front headers by short tubes. The upper ends of the back headers are connected to the steam drum by horizontal tubes.

The mud drum is a wrought steel box of square section, and is placed across the bottom of the front headers, which are connected to it by short tubes or nipples. This box being placed below the lowest part of the tubes collects any deposit or sediment. The blow-off cock is fitted to this chamber, and by its means any accumulated deposits can be blown out, or the boiler completely drained.

A handhole is provided opposite each tube in the header, by means of which the tubes can be examined, cleaned or renewed.

The circulation is good. The mixture of water and steam delivered into the steam drum impinges against baffle plates which separate the steam from the water. The steam and water drum is also fitted with wash plates to steady the water when the ship is rolling.

The Special Boiler Committee appointed by the British Parliament reported favourably on these boilers, and consequently a large number of battleships and large cruisers have been fitted with them. They are also largely used in the American Navy, and have given great satisfaction. The simplicity of the design, their cheapness, and the ease with which repairs can be accomplished will no doubt recommend their adoption in the Mercantile Marine to a much greater extent in the future than has been the case in the past, but Marine Engineers are not fully convinced

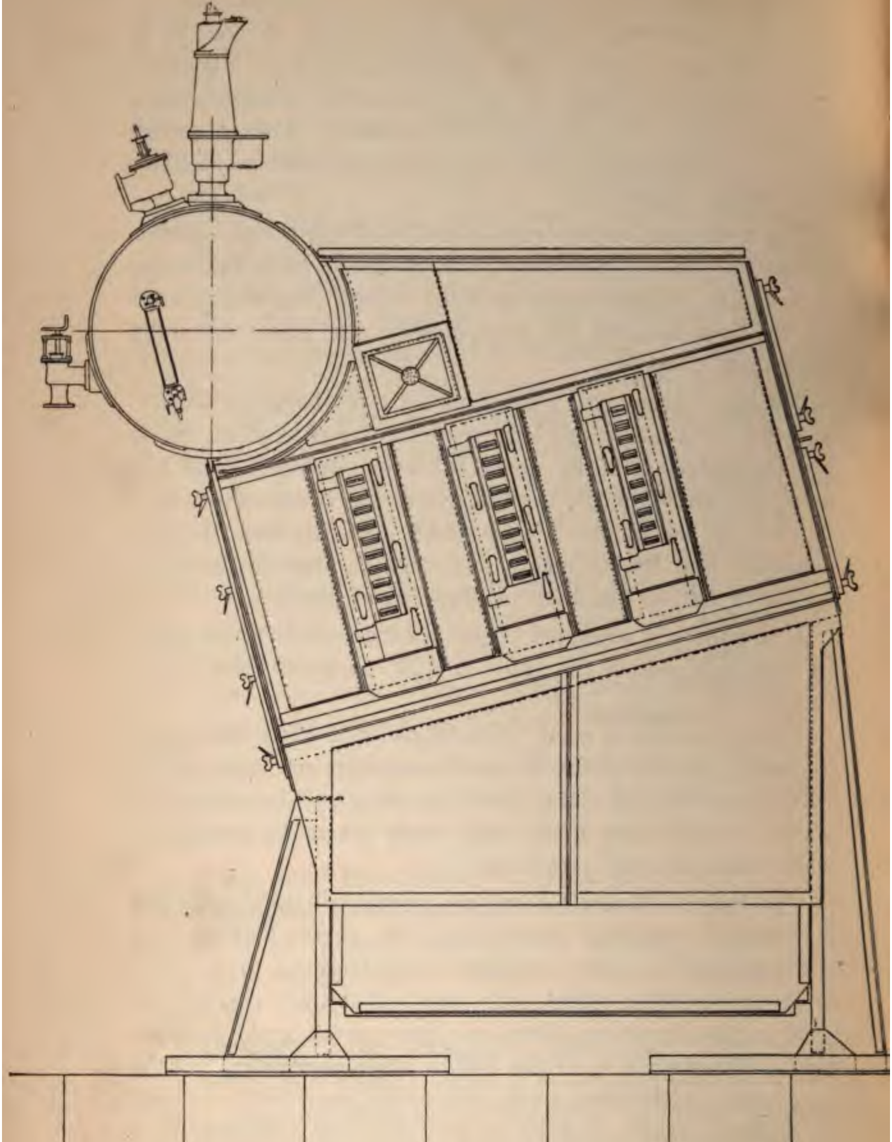


FIG. 46.—BABCOCK AND WILCOX BOILER.

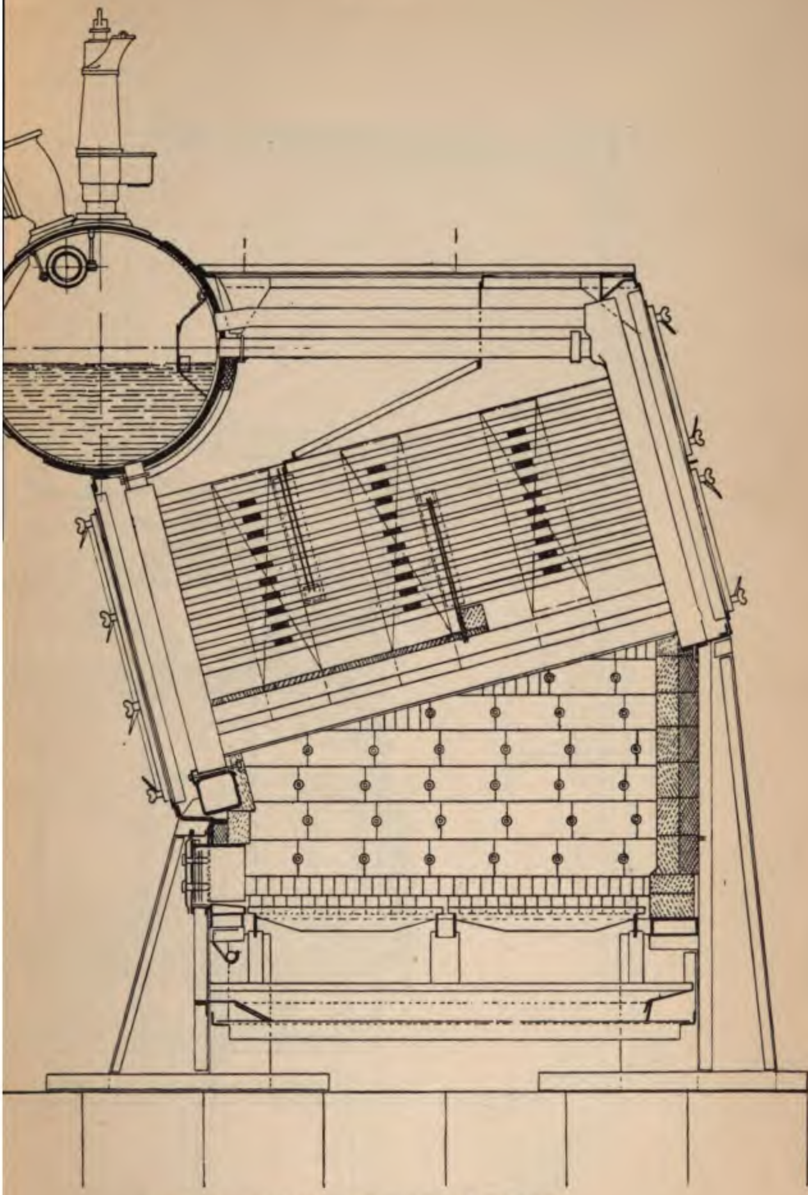


FIG. 47.—BABCOCK AND WILCOX BOILER.

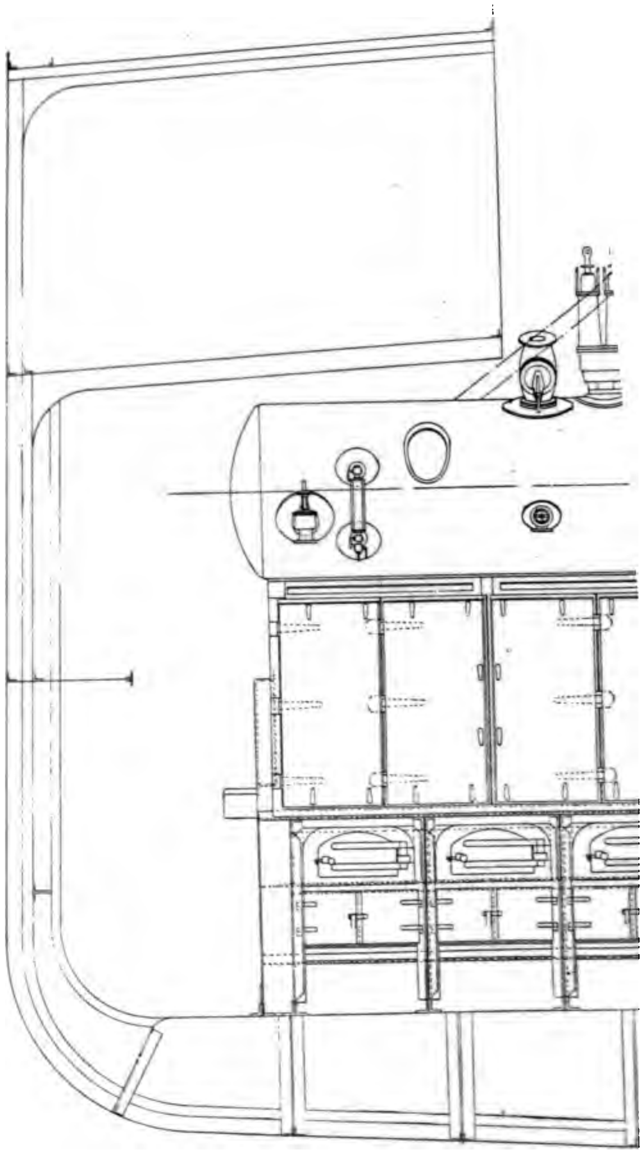
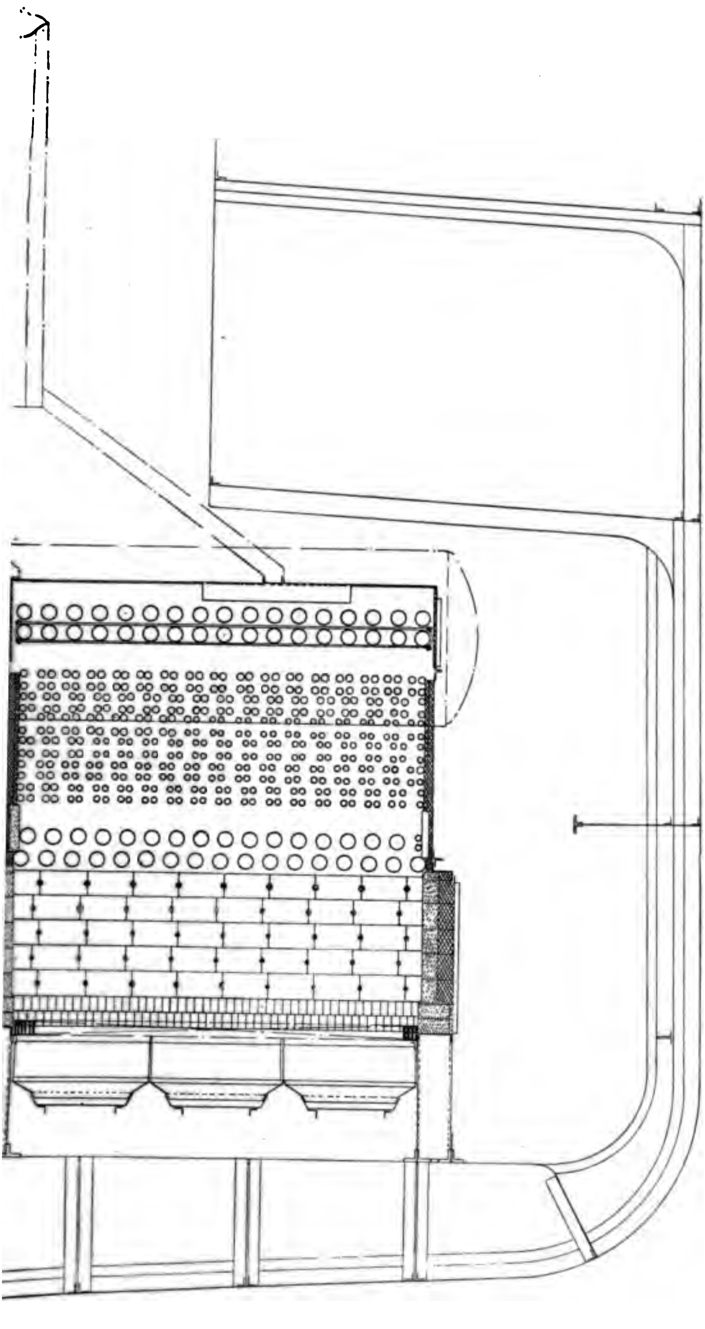


FIG.



ILCOX BOILER.

as to their length of life and efficiency as compared with an ordinary cylindrical boiler.

In the Marine type there are two designs, one having large and medium-sized tubes (known as mixed tube boilers), and the other having large tubes throughout. The mixed tube type gives a greater power for a given weight and space, but the large tube type finds most favour in the Mercantile Marine. The boiler illustrated by Figs. 46, 47, and 48 is of the mixed tube kind. It will be noticed that four small tubes take the place of one large one, and consequently give a greater heating surface in the same space.

YARROW BOILER. (Fig. 49.) This boiler, which is of the drowned tube class, has proved the most successful of the "Express" boilers.

It is one of the few recommended by the Boiler Committee to the Admiralty for further trials and use in large ships, but with larger tubes than are usually fitted in "Express" boilers.

STIRLING BOILER. (Fig. 50.) This boiler has given satisfaction to steam users on shore, and is now being employed for Marine work. The illustration shows the style used on board ship; for land work it is generally made with two bottom chambers. The feed water is delivered into the rear top chamber, the tubes connecting this with the bottom chamber acting as downcast pipes and feed water heater. Any sediment is deposited in the bottom chamber and blown out through the blow-off cock.

NICLAUSSE BOILER. (Figs. 51 and 52.) This boiler consists of a series of inclined double tubes, one inside the other, attached at the front end to vertical headers in such a manner that the colder water flows down the inside tube and returns to the front between the two tubes. The headers are made of cast steel or malleable cast iron, and are connected to the bottom of the steam drum by short neck pieces. A diaphragm in the headers completely separates the down-coming water from the ascending currents of hot water and steam. Each vertical row of tubes is attached to

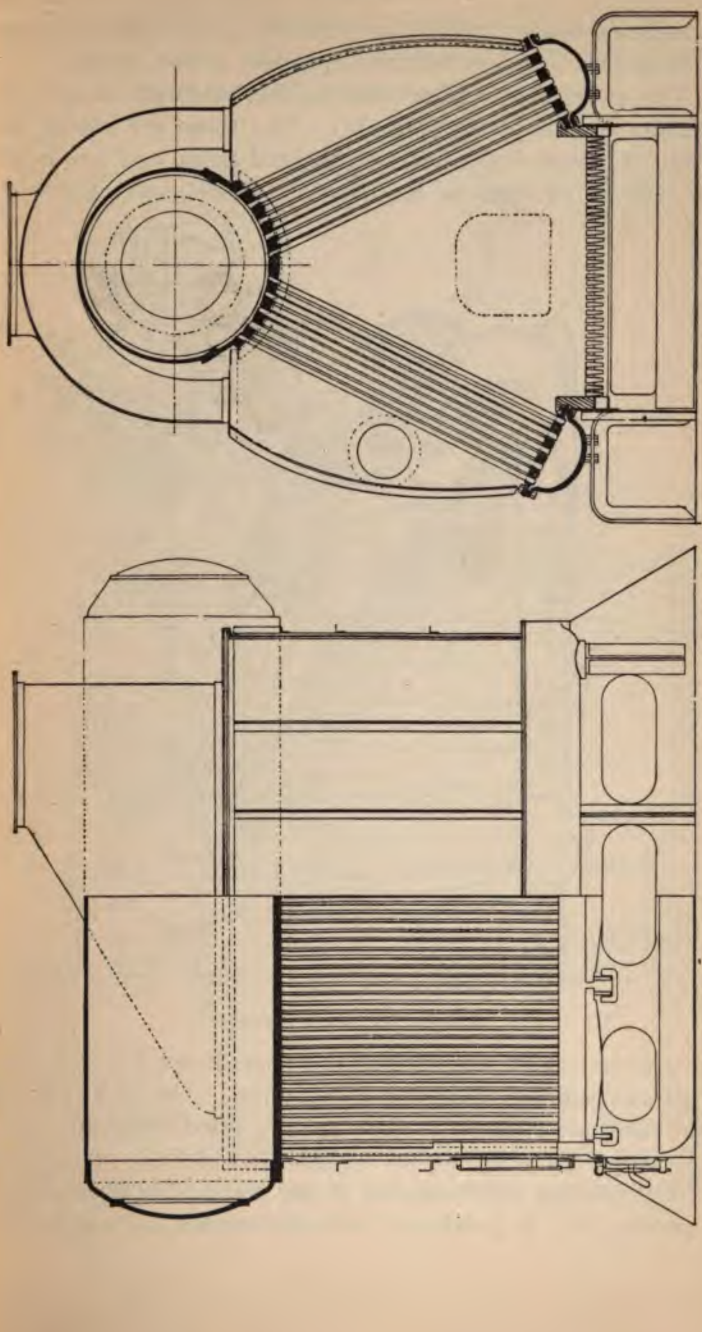


FIG. 49.—YARROW BOILER.

a separate header forming an element; the currents in each element are therefore quite independent of the others.

The attachment of the tubes to the headers is a special feature in this boiler (Fig. 53). The tubes are passed in from the front and fitted into tapered holes, and are kept in position by dogs on the outside. These joints are said

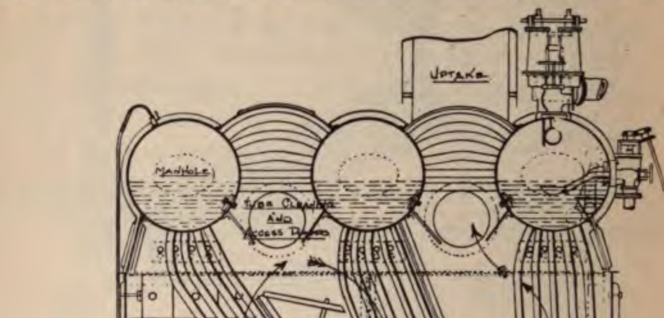


FIG. 50.—STIRLING BOILER.

to give no trouble, and a tube can be withdrawn for examination and replaced again in a few minutes. The back ends of the tubes are reduced in diameter, are fitted with caps, and are supported by a plate having holes in which the tubes rest.

The headers are connected at the bottom for blowing-off purposes, but it is evident the boiler cannot be completely

emptied except by taking out the tubes or removing the back end caps. This is certainly a great drawback, especially where the staff is limited.

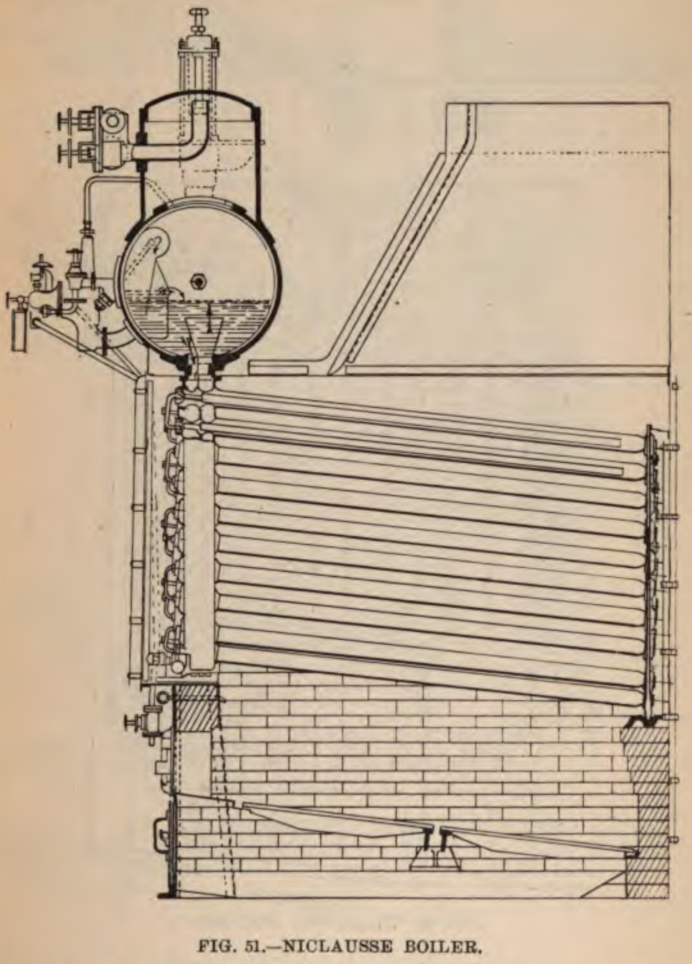


FIG. 51.—NICLAUSSE BOILER.

HEBBURN BOILER. (Figs. 54 and 55.) This boiler consists of two steam and water drums, and two mud drums connected by nearly vertical tubes and cross tubes slightly bent at the ends.

A special feature is the path of the hot gases, which is so arranged that every portion of the heating surface has to

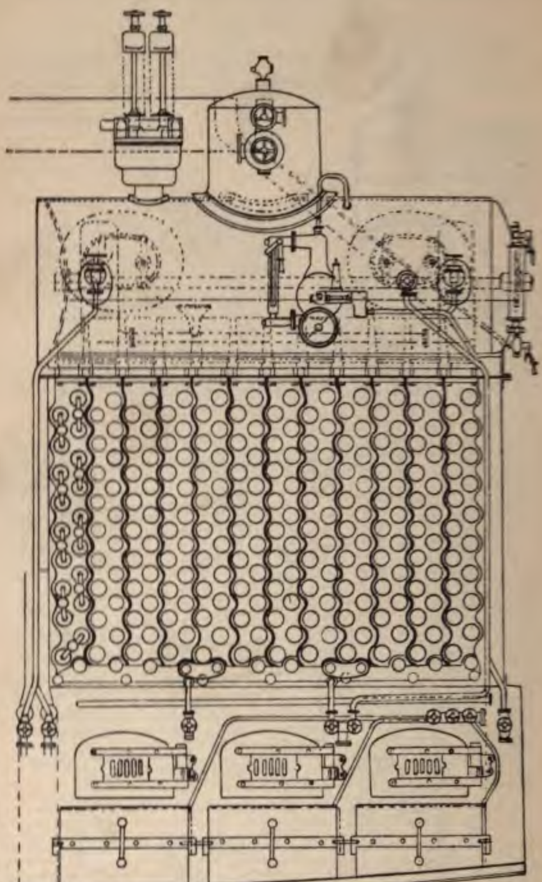
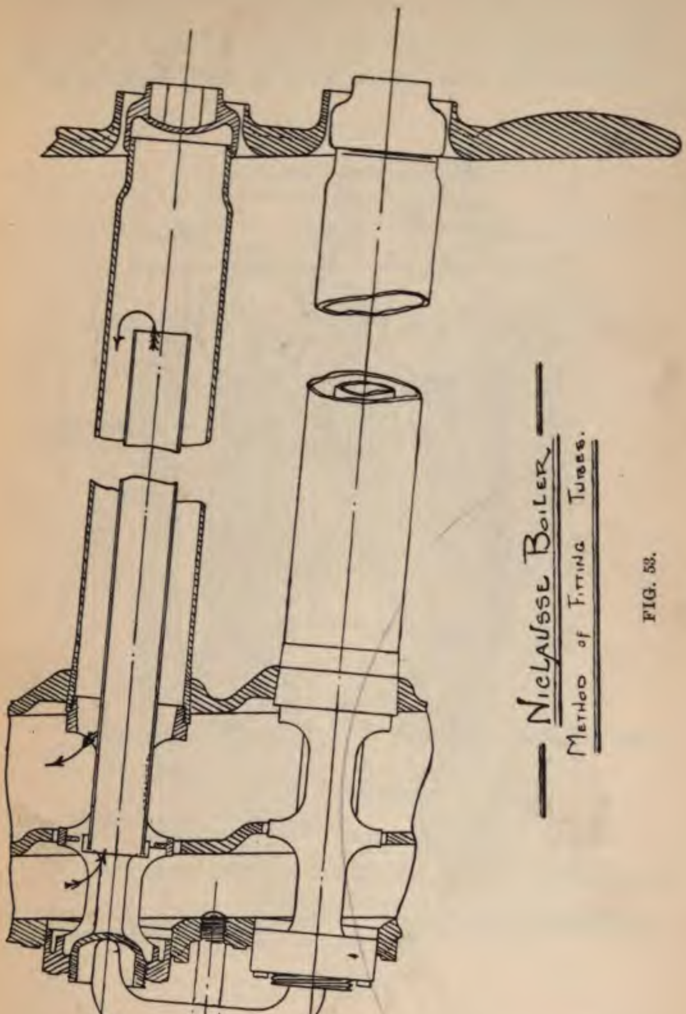


FIG. 52.—NICLAUSSE BOILER.

do its fair share of the work, and the combustion is said to be extremely good.

The feed water is pumped into the boiler at the rear end of the steam drums and descends through the back division



NICLAUSSE BOILER

METHOD OF FITTING TUBES.

FIG. 53.

of tubes, depositing any sediment it contains in the mud drum close to the blow-off cock.

The cross tubes between the steam and mud drums serve to maintain the same water level in both sections of the boiler.

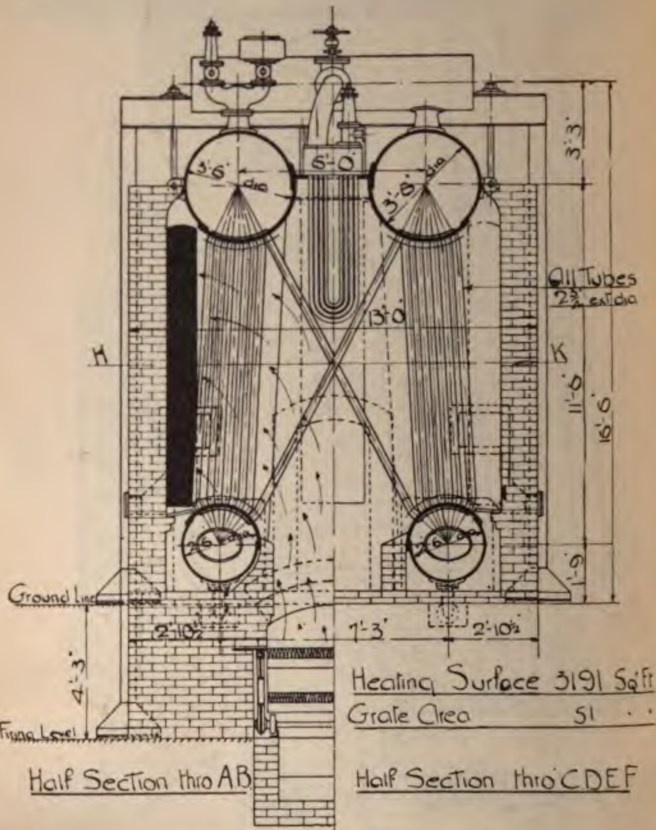


FIG. 54.—HEBBURN BOILER.

The boiler is extremely simple, and the steam trials have been highly satisfactory. It is amongst the youngest of the water tube family, and has not yet been adapted to suit Marine requirements.

In the foregoing pages boilermaking has been brought

right up to date, and the effort made to explain the principles of strength and construction in a plain practical manner may perhaps be of some use in assisting the engineer and boilermaker to understand, appreciate and apply what

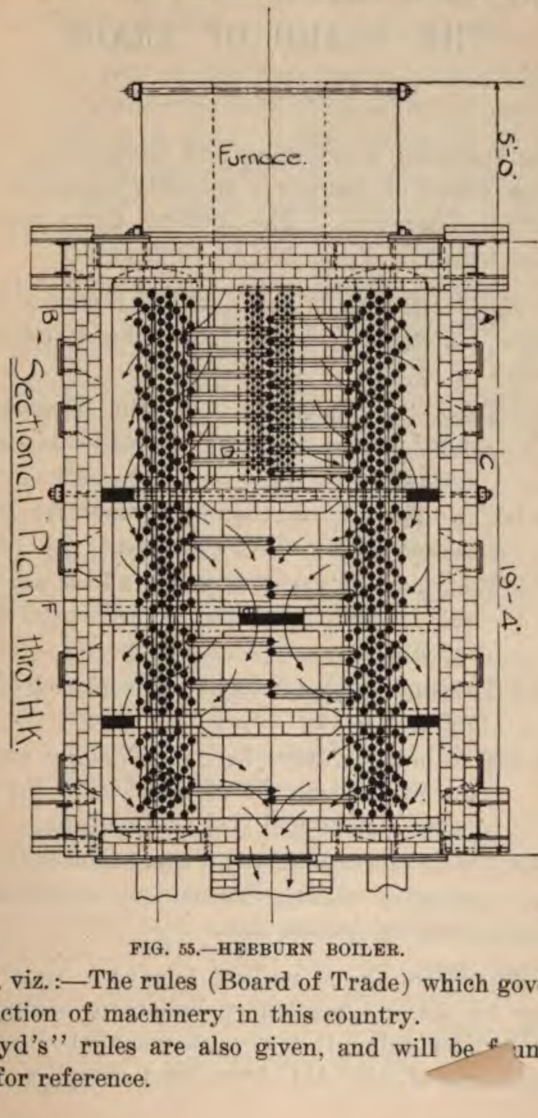


FIG. 55.—HEBBURN BOILER.

follows, viz.:—The rules (Board of Trade) which govern the construction of machinery in this country.

"Lloyd's" rules are also given, and will be found very useful for reference.

REGULATIONS AND SUGGESTIONS

ISSUED BY

THE BOARD OF TRADE.

The following Regulations and Suggestions issued by the Board of Trade are specially applicable for Engineer Surveyors. The numbers given are the official numbers of the respective clauses.

Surveyor's declaration.

3. The declaration of the Engineer Surveyor must contain statements of the following particulars, namely:—

(*a*.) That the machinery of the steamer is sufficient for the service intended, and in good condition;

(*b*.) The time (if less than one year) for which the machinery will be sufficient;

(*c*.) That the safety valves and fire hose are such and in such condition as are required by the Act;

(*d*.) The limit of the weight to be placed on the safety valves;

(*e*.) The limits (if any) beyond which, as regards the machinery, the steamer is, in the Surveyor's judgment, not fit to ply;

(*f*.) That the certificates of the engineer, or engineers, of the steamer are such as are required by the Act.

Responsibility of Surveyors.

4. Surveyors are in no case to give a declaration either for the hull, machinery, boilers, or equipments of a steamer unless they are satisfied that all the requirements of the Act have been complied with.

The Board of Trade will hold any Surveyor responsible to the fullest extent for the performance of the duty entrusted to him, and will support him in any reasonable steps he may think necessary for the full performance of his duty.

It is the duty of the Surveyor to satisfy himself personally regarding every detail of the survey of a vessel, and not to trust to any certificate or other document given by any person not responsible to the Board of Trade. This instruction does not, of course, apply to certificates of compass adjustment.

5. A Surveyor in the execution of his duties may go on board any steamship at all reasonable times, and inspect the same, or any part thereof, or any of the machinery, boats, equipment, or articles on board thereof, or any certificates of the masters, mates, or engineers, not unnecessarily detaining or delaying the ship from proceeding on any voyage; and if, in consequence of any accident to the ship, or for any other reason he considers it necessary, he may require the ship to be taken into dock for the purpose of surveying the hull. Powers of Surveyors.

If any person hinders a Surveyor from going on board any steamship or otherwise impedes him in the execution of his duties under the Act, he is liable to a penalty not exceeding £5. Any such case should be reported to the Board of Trade.

Surveyors are bound to make such returns as the Board of Trade require with respect to the build, dimensions, draught, burden, rate of sailing, room for fuel, and the nature and particulars of machinery of ships surveyed by them; and the owner, master and engineer of any ship so surveyed are bound, on demand, to give the Surveyors all such information and assistance as they require for the purpose of those returns.

Inspection to be made in the presence of owner, agent, or responsible person.

6. Surveyors should, if possible, make their inspection when the owner, agent, master, or chief mate and chief engineer of the vessel are present. Defects can then be pointed out to the proper persons without incurring the uncertainty and delay attendant upon messages delivered to subordinate officers. If the owners do not instruct a responsible representative to be present, the Surveyors will, if proper facilities have been given, proceed with the survey in his absence.

Written statement of repairs required to be given.

7. In order to prevent mistakes as to the nature and extent of repairs required by the Surveyors, whenever they cannot give a declaration until repairs are effected or alterations made, it is necessary that a written statement of the nature and extent of the defects to be remedied or alterations required, should in all cases be given to the owner or agent, whether applied for by him or not. A copy of the above should always be taken in the office press letter book.

If any difference of opinion should arise or any questions be raised between the Surveyor and the owner, manufacturers, or other person interested in the survey, the case should be reported to the Principal Officer.

Surveyor to complete survey where possible, but to sign only for details surveyed.

8. It is most desirable that a Surveyor should complete the survey of a vessel in every detail when he has commenced it, and, if possible, arrangements should always be made to secure this. In exceptional cases, however, when this cannot be arranged, one or more Surveyors should complete the parts of the survey left unfinished. The Surveyor who commenced the survey should not sign a declaration for any details which he has not surveyed, and for which consequently he is not prepared to hold himself personally responsible. He can only relieve himself from entire responsibility for every

detail, by distinctly noting on his declaration the points to which he has been unable to attend.

The Surveyor or Surveyors, who attend to the details of the survey which have been left unfinished, should also give declarations for the particulars they have attended to. The supplementary declaration should be attached to the main declaration.

Survey of ship belonging to another district.

9. If a vessel usually plying in one district is taken to another district to be surveyed, the Surveyor in the latter district is not to give a declaration without first communicating with the Surveyor in the former district, and, if there is any difference of opinion regarding the fitness of the vessel, the question should be referred to the Board of Trade.

Survey of vessels outside their plying districts.

The survey of steam ships for passenger certificates, other than foreign going or home trade, should not be completed in any district other than that for which the certificate is required, and in which the vessel is intended to ply. Persons who are interested in this regulation should be informed that a survey for such classes of certificates made out of the plying district, may not be sufficient for dispensing with another survey when the vessel arrives within such district, and for this further survey a fee of £1 will have to be paid notwithstanding the payment of fee for the previous survey.

This does not apply to the survey of the bottoms of these vessels in dry dock, or to inspections of them during construction, which may be performed at any port, provided the Surveyors forward the usual declaration to the Surveyors at the port at which the survey is to be completed, in order that it may be there connected with the main declaration.

Short declarations.

10. Declarations can only be granted for such time as the hull, equipments, boilers, and machinery will, in the opinion of the Surveyors, be sufficient for the service intended; but no declaration should be given

for less than three months in the case of a foreign-going steam ship, or for less than one month in the case of other passenger steam ships, without the special sanction of the Board of Trade. The only exceptions to this rule are the cases in which a certificate is required for the unexpired portion of the docking time,* or for the unexpired portion of the excursion season.

Incomplete declarations.

11. When a Surveyor is, from any reason whatever, unable to complete his declaration, an incomplete declaration should be sent to the person who applied for the survey, and it should contain a statement of the reasons why the declaration is incomplete. The Surveyor will take care to obliterate all items to which he is unable to certify.

Refusal of declarations.

12. A report should be sent to the Board of Trade of the name, port of registry, and official number of all vessels for which declarations are refused. This should be done as soon as possible after the vessel is withdrawn from survey, and full reasons for the refusal should be given.

The Surveyors will be supplied with a list of the vessels for which declarations have been refused, and care should be taken that no declaration is subsequently granted for any vessel contained in this list until the Surveyor has seen the papers relating to the case.

Survey of Hull.

Hulls to be examined in dry dock.

13. Surveyors are to determine whether the hulls of passenger vessels are in good condition and fit for the service for which they are intended; and they are to examine the hulls of such vessels outside and inside once at least in every year at such time as is most convenient to the owners.

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*The time for which the hull has been found to be sufficient after survey in dry dock.

Delays and inconveniences may to a great extent be prevented if notice is given by owners when a vessel, which requires a certificate, is undergoing repairs, or is in dry dock for any other purpose, as this notice may obviate the necessity of re-docking a vessel at an inconvenient time. Surveyors should explain this to owners or agents whenever an opportunity occurs.

Owner should give notice when a vessel is in dry dock.

14. Declarations should not be granted for steamers for the North Atlantic passenger trade, which have not hitherto been engaged in that trade, without first referring each case specially to the Board of Trade for instructions.

Declarations, North Atlantic trade.

15. The Board of Trade cannot regard as a satisfactory survey of a new steam ship a survey made only after the hull is complete, and is cemented and painted. Surveyors are therefore instructed to decline to grant declarations for any new steamer, unless they inspected her before she was painted or cemented. Any survey made while a vessel is being built is not to interfere with the complete inspection afterwards, but is to be made solely with a view of enabling the Surveyors to form an opinion of the workmanship, material, and construction.

New ships to be surveyed before being painted or cemented.

The bottoms of new vessels need not be examined in dry dock after launching, if they have been examined by the Surveyor before launching, unless he has special reasons for considering it necessary.

In cases in which Surveyors decline to grant declarations in consequence of not having inspected the vessel before the hull was painted and cemented, the owners or builders should be referred to the Board of Trade in order that the Board may decide in each case what steps should be taken.

Surveyors are sometimes called upon by owners to examine the bottoms of vessels during the night time. As such examinations cannot be considered

satisfactory Surveyors are instructed not to examine the bottoms of vessels except during daylight.

Collision bulkheads, water-tight compartment round stern tube, and other bulkheads.

16. An efficient and water-tight engine-room and stokehole bulkhead, as well as a collision water-tight bulkhead, and an after water-tight compartment to enclose the stern tube of each screw shaft, should be fitted in all sea-going steamers, both new and old, and in the absence of any of these the case must be specially referred to the Board of Trade before a declaration is given. As regards other bulkheads, although a thorough sub-division of the ship is desirable, the Surveyors should not for the present refuse to grant a declaration because these are not fitted.

The distance of the collision bulkhead from the after side of the stem measured at the level of the lower deck should not be less than one twentieth of the vessel's length measured from the after part of the stem to the fore part of the stern post, on the range of the upper deck beams, in one, two, and three decked and spar-decked vessels, but on the range of the main deck beams in awning-decked vessels.

If the collision bulkhead does not extend to the height of the upper or weather deck, full particulars should be reported to the Board before a declaration is issued.

The above regulation regarding the position and height of the collision bulkhead is intended to apply only to passenger steam ships coming under survey hereafter for the first time.

The collision bulkhead should not have any opening in it, nor should it be fitted with any valves or cocks for draining the compartment in front of it. Pipes should not be carried through it without the special permission of the Board of Trade.

In the case of vessels plying within smooth water

or partially smooth water limits, while it is desirable that the above requirements as to sea-going steamers should be complied with, the arrangement of bulkheads may be modified to meet the case of vessels of small size, or carrying only a small number of passengers. In all new iron or steel passenger steamers, however, with the exception stated below, an efficient collision bulkhead at least should be provided, and declarations should not be issued for vessels not so fitted without special instructions from the Board of Trade.

Steam launches intended exclusively for plying on very narrow waters will not be required to comply with the above regulation, but the intended limits should be notified for the Board's approval at an early stage of construction.

Screw tunnels and alternative arrangements.

17. In all sea-going screw passenger vessels there should be fitted in front of each stern tube stuffing box bulkhead either a complete water-tight tunnel extending from the stuffing box bulkhead to the after engine room bulkhead, or a water-tight compartment immediately before the stuffing box bulkhead, and extending forward of this not less than twelve times the diameter of the propeller shaft. In either case the tunnel or water-tight compartment should be substantially constructed with a stuffing box round the shaft at the bounding bulkhead through which it passes. The tunnel or compartment should be capable of confining the water which might enter the vessel in the event of any part of the aftermost stuffing box bulkhead or its fittings giving way. The height and width of the water-tight tunnel or compartment should be amply sufficient at every part to permit all work which may become necessary to the shaft couplings, bearings, &c., to be performed.

If there is an opening through the engine room

bulkhead, or the bulkhead forming the forward boundary of the water-tight compartment a properly constructed water-tight door, capable of being expeditiously opened and closed from a point well above the deep load line, preferably the upper deck, should be fitted to it. If no such opening is provided there should be fitted a strongly constructed, sufficiently roomy, and perfectly secure water-tight trunkway, extending to the upper deck, so that the interior of the water-tight tunnel or compartment may be at all times accessible.

The water-tight tunnels, compartments, and trunkways are in all cases of iron or steel vessels to be constructed of wrought iron or steel.

If there are manholes or openings in the floor of the tunnel care should be taken that all are efficiently fitted with covers or doors so arranged that they can easily and rapidly be made water-tight. Cocks or valves and pipes should be fitted to let the water off the floor of the tunnel or compartment, and should be made to open or shut from a point well above the deep load line.

Declarations may be granted for vessels requiring only partially smooth and smooth water certificates without water-tight tunnels.

Water-tight doors in bulkheads.

18. When inspecting steam ships, for which passenger certificates are required, the Surveyors should be careful to examine the water-tight doors, especially those in the stoke-hole bulkheads and coal bunkers. The Surveyors are not expected to insist on any particular description of door being fitted, nor to raise objections so long as they are satisfied that the arrangements provided are such as comply with the usual requirements for the service intended; at the same time suitably placed vertical doors, which can be opened and closed efficiently and expeditiously, would seem to be those on which most

reliance can be placed, more particularly for closing the communication with coal bunkers.

It is desirable that the closing edges of doors should be bevelled.

All doors on bulkheads required by the Regulations to be watertight should be capable of being worked from a point well above the deep load line, preferably the upper deck.

If the Surveyor is of opinion that there is danger of persons being injured while passing through a doorway in a bulkhead, owing to the door being of a quick closing type or for any other reason, a report of all the circumstances should be forwarded to the Board before a declaration is issued.

Periodical examination of the bottom outside.

19. In no case should a Surveyor grant a declaration for a steamer, the outside of the bottom of which has not been examined by a Board of Trade Surveyor during the 12 months preceding the date of survey, and in no case should a declaration be granted for a longer period than a period terminating within 12 months from the date of the last outside examination. For instance, if a steamer be examined outside on the 1st of January, she will be entitled to a certificate for 12 months, *i.e.*, until the 1st of January following. But supposing the survey is not completed until the 1st of February, the Surveyor is, if the vessel be fit, to grant his declaration for a period ending the 1st of January following, and no longer, unless the vessel be re-docked for examination. Surveyors should not depart from this instruction in any case without the written directions of the Board of Trade.

Examination of bottom, cocks, &c., in dry dock.

20. Before granting a declaration for 12 months, Surveyors should make a thorough examination of the outside of the bottom of the ship in a dry dock, after it has been cleaned and before it is painted. They should examine the pintles and braces of the

rudder, as well as the bearings of the screw shaft and the different openings, valves, bolt-heads, &c., in the ship's bottom. All bolts connecting cocks, valves, or pipes to the skin of the ship should have the heads outside, and such heads should be either countersunk or cupheaded. All discharge valves and sea cocks should be taken out for examination while the vessel is in dry dock.

Before Surveyors finally pass the hull of a vessel they are to see it fit for going into the water.

Ceiling of all ships to be removed as Surveyor thinks necessary.

21. For examining the internal parts of all ships, they should have parts of the ceiling removed in order that they may ascertain the condition of the ship; they should also carefully examine the cement to ascertain its condition, and where they find it cracked or doubtful, they should extend their examination, and cause such a portion of the cement to be removed, as may in their judgment be necessary to enable them to form a correct opinion as to the state of the ship where covered by the cement.

The roses for the deck pumps, also those for the bilge pumps, should be taken out of the bilges and cleaned, and the sluices and cocks on the bulkheads, if any, inspected and put in good working order.

Examination of vessel in engine room, coal bunkers, &c.

The Surveyors should examine very particularly all parts of the ship (*i.e.*, plating, stringers, frames, doors, &c.), in the engine room, boiler space, and coal bunkers, the bunkers, of course, being empty.

Entries in docking book.

22. The date of the last outside inspection of the bottom is to be entered in the docking book, as it is from that date that the docking time is to count. If the hull is not passed at all, or not passed for the maximum time (12 months), the fact should be noted, both in the docking book and also in the declaration for the vessel. Vessels of which the names are entered in the docking book, with no

notes to the contrary, will be considered to have been passed for 12 months, and the Surveyors who inspected them will be held responsible for passing them for that period.

The repairs which have been executed from time to time should be recorded in the office docking book.

23. Any questions of doubt as to the strength of vessels should be referred to the Board of Trade, and in future midship sections are to be submitted, at as early a date as possible, in the case of all new steam ships building under survey for which passenger certificates are required, unless the Surveyor has satisfied himself that the scantlings are, in the case of iron or steel vessels, in all respects at least equivalent to the standard laid down in the Tables of Freeboard. Strength of vessel.

In the case of vessels of such large dimensions that the "numerals" are in excess of those given in the Rules of the Corporations for the survey and registry of shipping approved by the Board under section 443 of the Merchant Shipping Act, 1894, and in the case of vessels of which the length is greater than 16 times the depth, midship sections are to be submitted.

In the case of vessels already holding passenger certificates for certain specified limits, Surveyors should not issue declarations for more extended limits without first obtaining the sanction of the Board of Trade.

When a vessel which has been surveyed for a passenger certificate is not in every respect in good condition, although the defects may not be sufficient to warrant the withholding of the required declaration, and although the vessel may be practically fit for the service intended, the Surveyor should, when he grants the declaration, forward to the Board of

Trade a report showing the nature of the defects in question.

It is advisable that sketches of the midship sections of all vessels entered therein be made in the docking book.

In all cases in which passenger steamers are intended to be fitted with shoots, for ashes or galley refuse, constructed of metal other than wrought iron or steel, full particulars with drawings to scale should be obtained by the Surveyor before any part of the work is commenced, and at once submitted to the Board of Trade for consideration.

Coamings, skylights, scuppers, ports, gratings, &c.

24. The height of the coamings around the openings in the weather deck, and the means provided for securely protecting or fastening down the skylights, hatches, bunker lids, &c., as well as the sufficiency of the water ports and scuppers for relieving that deck of water, are important items to be noted by the Surveyors in any sea-going vessel coming under their survey. All openings in the weather deck of a sea-going steam or sailing vessel carrying passengers or emigrants, whether such weather deck be a poop, awning deck, spar deck, promenade deck, or forecastle, should be fitted with covers so placed and arranged that they may be expeditiously shipped and the deck made watertight.

The openings which are over the stoke-holes and around the funnels and engine-room skylights of steam vessels should be fitted with gratings, as well as with iron or steel covers, having suitable means for securing them in position from the weather deck. The coamings of such openings should be of ample strength and height.

The other openings in the weather deck should be fitted with gratings or hatch covers and tar-

paulins, and also with means for effectually securing the tarpaulins and making the openings water-tight.

All openings in the main or lower deck should also be fitted with gratings or hatch covers and tarpaulins, or other means for making the openings water-tight.

Each cover, hatch-grating, &c., should be kept and secured in a suitable place, at all times accessible and near to the opening for which it is intended.

The covers and coamings of cargo hatchways on the weather deck should be water-tight and have ample strength and height.

Many casualties have happened through the engine-room skylights, coamings, companions, &c., being of light construction and insufficiently secured, instead of being of sufficient strength to form permanent parts of the structure of the ship; casualties have also arisen through the front of poops and bridge-houses being weak and insufficiently secured. The fore and aft beams, or "strong backs," in the hatchways, and the hatches themselves have often been found too weak to bear the pressure of water which finds its way on to them in a heavy sea.

The Surveyors should, in granting declarations for passenger steamers, look especially to these points, and if they have any doubt should call the attention of the Principal Officer to the subject, and should not grant a declaration without his authority. The Surveyors should in like manner in the case of ships other than passenger steamers, look to these amongst other points when reporting a ship for detention as unsafe.

Sidelights and frames and deadlights.

25. Sidelights in any part of the vessel which might become flooded with water, if one of such sidelights were broken, must be fitted with deadlights; but this requirement need not be enforced in respect to sidelights in erections above the

weather-deck, provided that all deck openings are properly protected. In the case of vessels carrying limited home trade or excursion certificates, sidelights in full poops or forecastles, or between decks immediately under awning or promenade decks may be exempt, except in cases where the Surveyor is of opinion that the glass in the sidelight is insufficiently substantial, or that the breaking of the sidelights in such spaces would endanger the safety of the vessel. Vessels now carrying passenger certificates should not be interfered with, provided the spirit of the Board's regulation is complied with. Sidelights in the exempt spaces mentioned above should, however, be fitted with deadlights when such spaces are intended to be so filled with cargo as to render access to the sidelights difficult when the vessel is at sea. All stern and other ports must be fitted in such a manner that they can be quickly and efficiently secured. Sidelights should not be passed in which the fittings and framework are made of cast iron, in any steam ship coming under survey for the first time. This regulation applies to the framework, &c., and not to the deadlights or storm shutters, and it need not be enforced in the case of any vessel not exceeding 50 tons net register and plying exclusively within Stm. 5 limits.

Iron decks to be sheathed in winter.

26. During the winter months from 1st September to 30th April, vessels with iron decks should not be passed for Stm. 1, 2, or 3 certificates unless the decks are sheathed with wood.

Vessels propelled by electricity or other mechanical power.

27. When application is made for survey for passenger certificate of a vessel propelled by electricity or other mechanical power, the Surveyor should be guided as to the survey of the hull and equipments by the Board's printed and other regulations with respect to the survey of steam ships. If the propelling machinery or any portion of its

accessories is such as will, in the Surveyor's judgment, injuriously affect the hull or equipments, or any portion thereof, the Surveyor should report fully to the Board regarding the effects anticipated, as well as to the means, if any, for preventing them which the owner is willing to adopt.

With regard to the propelling machinery employed (including accessories), the Surveyor should, before issuing his declaration, report fully to the Board as to the principles involved in its construction, and as to the exceptional dangers, if any, which would, in the Surveyor's judgment, attend its use, such plans being appended to the report as may be necessary to make it intelligible. The estimated speed of the vessel in knots per hour, the number of revolutions of the propelling shaft, and the probable time which the machinery is capable of maintaining that speed, should be included in the report.

The issue of the statutory declarations for vessels propelled in any manner not contemplated by the Board's present printed regulations, should in all cases be withheld until the sanction of the Board of Trade has been obtained.

Boilers and Superheaters.

80. Surveyors are required by the Act to fix the limits of weight to be placed on the safety valves of passenger steam ships and to determine whether the boiler and machinery are sufficient for the service intended and in good condition. The Surveyor to fix pressures on safety valves.

81. The Surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, and according to the workmanship and material. The Board of Trade have arranged to receive, for examination by their Surveyors, plans and particulars of boilers before the commencement of manufacture, by these means Working pressure to be fixed by calculation.

hoping to prevent questions arising after the boilers are finished and on board. This practice has been found to work well in saving time to the Surveyors, and in preventing expense, inconvenience, and delay to owners. The senior Engineer Surveyors should therefore receive and report on any plans of boilers intended for passenger vessels that may be submitted in due course with the Form Surveys 6. They are not to report on any tracing or plan that is not accompanied by that form. When the Surveyor has received plans and tracings of new boilers, or of alterations to boilers, and has approved of them, he will, of course, be careful in making his examination from time to time to see that they are followed in construction. When he has not had the plans submitted, but is called in to survey a boiler, he will measure the parts, note the details of construction, and, if necessary, bore the plates to ascertain their thickness, &c., before he gives his declaration. And in the event of any novelty in construction, or of any departure from the practice of staying and strengthening noted in these Regulations, or of the boiler being made of steel, he should report full particulars to the Board of Trade before fixing the working pressure.

The thickness of plates used in the construction of boilers should not be less than $\frac{5}{16}$ in.

The Surveyor cannot declare a boiler to be safe unless he is fully informed as to its construction, material, and workmanship. He should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called in to survey until after it is completed, and fixed in the ship.

Iron Boilers.

Respecting stays.

82. In the case of new boilers the Surveyor may allow a stress not exceeding 7000lbs. per square inch

of net section on solid iron screwed stays supporting flat surfaces, but the stress should not exceed 5000 lbs. when the stays have been welded.

A stress of 6000lbs. may be allowed on the net section of iron stay tubes, providing that the net thickness is in no case less than $\frac{1}{4}$in.

When the threads of longitudinal stays are finer than six per inch, the depth of the external nuts should be at least $1\frac{1}{4}$ times the diameter of the stay.

The areas of diagonal stays are found in the following way:—Find the area of a direct stay needed to support the surface, multiply this area by the length of the diagonal stay, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the diagonal stay; the quotient will be the area of the diagonal stay required.

When gusset stays are used, their area should be in excess of that found in the above way.

83. When the tops of combustion boxes or other parts of a boiler are supported by solid girders of rectangular section, the following formula should be used for finding the working pressure to be allowed for the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers that the ends are properly bedded to the edges of the tube plate and the back plate of the combustion box:— Girders for flat surfaces.

$$\frac{C \times d^2 \times T}{(W - P)\, D \times L} = \text{Working pressure.}$$

W = Width of combustion box in inches.

P = Pitch of supporting bolts in inches.

D = Distance between the girders from centre to centre in inches.

L = Length of girder in feet.

d = Depth of girder in inches.

T = Thickness of girder in inches.

N = Number of supporting bolts.

$C = \frac{N \times 1200}{N+1}$ when the number of bolts is odd.

$C = \frac{(N + 1)\ 1200}{N+2}$ when the number of bolts is even.

The working pressure for the supporting bolts and for the plate between them should be determined by the rules for ordinary stays and plates.

Flat surfaces of boilers.

84. The pressure on plates forming flat surfaces is found by the following formula:—

$$\frac{C \times (T + 1)^2}{S - 6} = \text{Working pressure.}$$

T = Thickness of the plate in sixteenths of an inch.

S = Surface supported in square inches.

C = Constant according to the following circumstances.

C = 192 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and doubling strips not less in width than two-thirds the pitch of the stays and of the thickness of the plates, are securely riveted to the outside of the plates they cover.

C = 168 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers not less in diameter than two-thirds the pitch of the stays and of the same thickness as the plates, securely riveted to the outside of the plates they cover.

C = 132 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts on both sides of the plates, and with washers outside the plates at least three times the diameter of the stay, and two-thirds the thickness of the plates they cover.

C = 120 when the plates are not exposed to the impact of heat or flame and the stays are fitted with nuts on both sides of the plates.

C = 90 when tube plates are not exposed to the direct impact of heat or flame, and the stays are fitted with nuts.

C = 70 when tube plates are not exposed to the direct impact of heat or flame and the stay tubes are screwed and expanded.

C = 70 when the plates are not exposed to the impact of heat or flame, and the stays are screwed into the plates and riveted over.

C = 60 when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, and the stays fitted with nuts and washers, the latter being at least three times the diameter of the stay, and two-thirds the thickness of the plates they cover.

C = 54 when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, and the stays fitted with nuts only.

C = 80 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plates and fitted with nuts.

C = 60 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the

plates, and having the ends riveted over to form substantial heads.

C = 36 when the plates are exposed to the impact of heat or flame, with steam in contact with the plates, with the stays screwed into the plates and having the ends rivited over to form substantial heads.

When the plates are not exposed to the impact of heat or flame and doubling plates covering the whole of the flat surfaces are riveted to the plates, the working pressure may be found by the following formula:—

$$\frac{C\ (T+1)^2 + C\ (T_1+1)^2}{S-6} = \text{Working pressure.}$$

T = Thickness of the plate in sixteenths of an inch.

T_1 = Thickness of the doubling plate in sixteenths of an inch.

S = Surface supported in square inches.

C = Constant applicable to the case as given above.

When doubling plates do not cover the whole of the flat surfaces but are fitted between the rows of supporting stays the strength allowed for them should be two-thirds only of that which would be allowed for similar doubling plates extending beyond and embracing the supporting stays.

In calculating the working pressure of the portion of tubes plates between the boxes of tubes, the value of S in the above formula should be found as follows:—

$$\frac{D^2 + d^2}{2} = S$$

Where D = the horizontal pitch of the stay tubes in inches.

and d = the vertical do. do. do.

The pitches should be measured from centre to centre of the stay tubes and no deduction should be made for any tubes in the contained surface.

In the body of tube plates the value of S may be found in the ordinary way and the area of the tubes in the space bounded by the stay tubes may be deducted.

In cases where plates are stiffened by T or L irons, and a greater pressure is required for the plates than is allowed by the use of the above constants, the case should be submitted for the consideration of the Board of Trade.

When a circular flat end is bolted or riveted to an outside ring or flange of a cylindrical shell, S in the formula may be taken as the area of the square inscribed in the circle passing through the centres of the bolts or rivets securing the end, provided the ring or flange is of sufficient thickness.

When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the Surveyor must act according to the circumstances that present themselves at the time of the survey, and it is expected that in cases where the riveted ends of screwed stays in the combustion boxes and furnaces are found in this state it will be often necessary to reduce the constant 60 to about 36.

85. The Surveyors should not in any case allow a greater compressive stress on the tube plates than 11000lbs., which is that used in the following formula:— **Compressive stress on tube plates.**

$$\frac{(D-d)\ T \times 22000}{W \times D.} = \text{Working pressure.}$$

D = least horizontal distance between centres of tubes in inches.

d = inside diameter of ordinary tubes in inches.

T = thickness of tube plate in inches.

W = width of combustion box in inches between the tube plate and back of fire box, or distance between the combustion box tube plates when the boiler is double ended and the box common to the furnaces at both ends.

Wet-bottomed boilers to be periodically lifted from their seats.

86. Having regard to many cases in which serious defects have been discovered, the Surveyors should take care that wet-bottomed and other boilers, the outside of the bottom of which cannot be seen, are lifted for inspection at least once in every four years, or oftener if the Surveyors consider it necessary. It will often be found necessary long before this to reduce the pressure, unless the boilers are lifted from their seats to enable the Surveyors to judge of their efficiency. If the owners in any special case have any good reasons for not wishing to lift them when the Surveyor requires it, the Surveyor should submit the whole case in detail to the Board of Trade for their consideration. The Surveyor must recollect that he is not to certify as sufficient any boiler respecting which he cannot thoroughly satisfy himself.

The Surveyor should record in the hydraulic test book the dates on which boilers which have been lifted are inspected, and whether the boilers were out of the ship when examined, and if out of the ship where they were examined. Boilers which have been lifted should in all cases be subjected to the hydraulic test before they are re-set.

Cylindrical boilers.

87. The Board of Trade consider that boilers well constructed, well designed, and made of good material should be allowed an advantage in the matter of working pressure over boilers inferior in any of the above respects, as unless this is done the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged.

They have therefore caused the following rules to be prepared:—

When the cylindrical shells of boilers are made of the best material with all the rivet holes drilled in place and all the seams fitted with double butt straps each of at least five-eighths the thickness of the plates they cover, and all the seams at least double riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of construction, then 5 may be used as the factor of safety. The tensile strength of the iron is to be taken as equal to 47000 lbs. per square inch with the grain and 40000lbs. across the grain. If, however, the iron be tested and the elongation measured in a length of 10 inches is not less than 14 per cent. with, and 8 per cent. across the grain, and the Surveyors are otherwise satisfied as to the quality of the plates and rivets, 4·5 may be used as the factor of safety instead of 5, in which case the minimum actual tensile strength of the plates should be used in calculating the working pressure.

When the above conditions are not complied with, the additions in the following scale should be made to the factor of safety, according to the circumstances of each case:—

A†	·15	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place after bending.
B†	·3	To be added when all the holes are fair and good in the longitudinal seams, but drilled before bending.
C	·3	To be added when all the holes are fair and good in the longitudinal seams, but punched after bending.
D	·5	To be added when all the holes are fair and good in the longitudinal seams, but punched before bending.
E*	·75	To be added when all the holes are not fair and good in the longitudinal seams.

F	·1	To be added if the holes are all fair and good in the circumferential seams, but drilled out of place after bending.
G†	·15	To be added if the holes are fair and good in the circumferential seams, but punched before bending.
H	·15	To be added if the holes are fair and good in the circumferential seams, but punched after bending.
I†	·2	To be added if the holes are fair and good in the circumferential seams, but punched before bending.
J*	·2	To be added if the holes are not fair and good in the circumferential seams.
K	·2	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and double riveted.
L	·1	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and treble riveted.
M	·3	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are double riveted.
N	·15	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are treble riveted.
O	1·0	To be added when any description of joint in the longitudinal seams is single riveted.
P	·1	To be added if the circumferential seams are fitted with single butt straps and are double riveted.
Q	·2	To be added if the circumferential seams are fitted with single butt straps and are single riveted.
R	·1	To be added if the circumferential seams are fitted with double butt straps and are single riveted.
S†	·1	To be added if the circumferential seams are lap and are double riveted.
T	·2	To be added if the circumferential seams are lap and are single riveted.
U	·25	To be added when the circumferential seams are lap and the strakes of plates are not entirely under or over.
V‡	·3	To be added when the boiler is of such a length as to fire from both ends, or is of unusual length, as in the case of flue boilers and the circumferential seams fitted as described opposite P, R, and S, but, when the circumferential seams are as described opposite Q and T, V·4 should be added.

W*	·4	To be added if the longitudinal seams are not properly crossed.
X*	·4	To be added when the iron is in any way doubtful, and the Surveyor is not satisfied that it is of the best quality.
Y††	1·65	To be added if the boiler is not open to inspection during the whole period of its construction.

When marked * the factor may be increased still further if the workmanship or material is such as in the Surveyor's judgment renders such increase necessary.

† When the holes are to be rimered or bored out in place the case should be submitted to the Board as to the reduction or omission of A, B, G, and I, as heretofore.

‡ When the middle circumferential seams are double strapped and double riveted or lap and treble riveted, and the calculated strength not less than 65 per cent. of the solid plate S ·1 and V ·3 may be omitted. The end circumferential seams in such cases should be at least double riveted.

†† When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factors to be used.

The strength of ordinary joints is found by the following method:—

$$\frac{(\text{Pitch}-\text{Diameter of rivet}) \times 100}{\text{Pitch.}} = \text{Percentage of}$$

strength of plate at joint as compared with the solid plate.**

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{Pitch} \times \text{thickness of plate.}} = \text{Per-}$$

centage of strength of rivets as compared with the solid plate.§

Taking iron as equal to 47000lbs.* per square inch and using the smaller of the two per-centages as the

**For the maximum pitch of rivets *see* Appendix A, page 307.

§ If the rivets are exposed to double shear, multiply the percentage as found by 1·75.

*Or the minimum tensile strength if it has been tested.

strength of the joint and adopting the factor of safety as found from the preceding scale then:—

$$\frac{47{,}000^* \times \text{least \% strength of joint} \times 2 \times \text{plate thickness in ins.}}{\text{Inside dia. of boiler in ins.} \times \text{factor of safety} \times 100} =$$

Pressure to be allowed per square inch on the safety valves. (See the formulæ as given in detail in Appendix A.)

Riveting.

In the case of ordinary zig-zag riveting the strength through the plate diagonally between the rivets is equal to that horizontally between the rivets, when diagonal pitch $= \frac{6}{10}$ horizontal pitch $+ \frac{4}{10}$ diameter of rivet.

Plates that are drilled in place should be taken apart and the burr taken off, and the holes slightly countersunk from the outside.

Butt straps, &c.

Butt straps should be cut from plates and not from bars, and should be of as good a quality as the shell-plates, and those for the longitudinal seams should be cut across the fibre. When the straps are drilled in place they should be taken apart, the burr taken off, and the holes slightly countersunk from the outside.

When single butt straps are used, they should be one-eighth thicker than the plates they cover.

The diameter of the rivets should in no case be less than the thickness of the plates of which the shell is made, but it will be found when the plates are thin, or when lap-joints or single butt straps are adopted, that the diameter of the rivets should be in excess of the thickness of the plates.

Formulæ for the riveted joints and maximum pitches of rivets and also diagrams of different descriptions of riveted joints will be found in Appendix A.

*Or the minimum tensile strength if it has been tested.

Dished ends, unless of the thickness required for a flat end, should be stayed; but when they are equal to the pressure needed, when considered as portions of spheres, the stays, when solid, may have a stress of 14000lbs. per square inch of net section, but the stress should not exceed 10000lbs. when the stays have been welded, and such stays should be properly distributed. If dished ends are not equal to the pressure needed when considered as portions of spheres they should be stayed as flat surfaces. Stays for dished ends.

Hemispherical ends subjected to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness. The ends should be formed of not less than four pieces. Hemispherical ends, manholes, doors and domes.

Compensating rings of at least the same effective sectional area as the plate cut out, should be fitted round all manholes and openings, and in no case should the rings be less in thickness than the plates to which they are attached. The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally. It is very desirable that the compensating rings round openings in flat surfaces should be made of L or T iron. Cast-iron doors should not be passed.

The neutral part of the boiler shells under steam domes should be efficiently stiffened and stayed, as serious accidents have arisen from the want of such precautions.

The boilers should be tested by hydraulic pressure to twice the working pressure in the presence, and to the satisfaction, of the Board's Surveyors. (*See* Clause 115.) Hydraulic test.

Circular furnaces.

88. Circular furnaces with the longitudinal joints welded or made with single butt straps double riveted, or double butt straps single riveted:—

$$\frac{90000 \times \text{the sq. of the thickness of the plate in inches}}{(\text{Length in feet} + 1) \times \text{diameter in inches.}}$$

= working pressure per square inch, provided it does *not* exceed that found by the following formulæ:—

$$\frac{9000 \times \text{thickness in inches}}{\text{diameter in inches.}} = \left\{ \begin{array}{l} \text{Working pressure} \\ \text{per square inch.} \end{array} \right.$$

The second formulæ limits the crushing stress on the material to 4500lbs. per square inch.

The length to be measured between the rings if the furnace is made with rings.

If the longitudinal joints instead of being butted are lap-jointed in the ordinary way and double riveted, then 75000 should be used instead of 90000, but where the lap is bevelled and so made as to give the flues the form of a *true* circle, then 80000 may be used.

When the material or the workmanship is not of the best quality, the constants given above should be reduced, that is to say, the 90000 may become 80000; the 80000 may become 70000; the 70000 may become 60000; when the material and the workmanship are not of the best quality, such constants may require to be further reduced, according to circumstances and the judgment of the Surveyor. Some of the conditions of best workmanship are that the rivet holes shall be drilled after the bending is done and when the plates are in place, and the plates afterwards taken apart, the burr on the holes taken

off, and the holes slightly countersunk from the outside.*

89. The strength of the joints of cylindrical super- Cylindrical superheaters.
heaters and the factor of safety are found in a similar manner as for cylindrical boilers and steam receivers, but instead of using 47000lbs. as the tensile strength of iron 30000lbs. is adopted, unless where the heat or flame impinges at, or nearly at, right angles to the plate, then 22400lbs. is substituted.

When a superheater is constructed with a tube subject to external pressure the working pressure should be ascertained by the rules given for circular

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*The following examples serve to show the application of the constants for the different cases that may arise:

Furnaces with butt joints and drilled rivet holes.	90000 where the longitudinal seams are welded. 90000 where the longitudinal seams are double riveted and fitted with single butt straps. 80000 where the longitudinal seams are single riveted and fitted with single butt straps. 90000 where the longitudinal seams are single riveted and fitted with double butt straps.
Furnaces with butt joints and punched rivet holes.	85000 where the longitudinal seams are double riveted and fitted with single butt straps. 75000 where the longitudinal seams are single riveted and fitted with single butt straps. 85000 where the longitudinal seams are single riveted and fitted with double butt straps.
Furnaces with lapped joints and drilled rivet holes.	80000 where the longitudinal seams are double riveted and bevelled. 75000 where the longitudinal seams are double riveted and not bevelled. 70000 where the longitudinal seams are single riveted and bevelled. 65000 where the longitudinal seams are single riveted and not bevelled.
Furnaces with lapped joints and punched rivet holes.	75000 where the longitudinal seams are double riveted and bevelled. 70000 where the longitudinal seams are double riveted and not bevelled. 65000 where the longitudinal seams are single riveted and bevelled. 60000 where the longitudinal seams are single riveted and not bevelled.

In the case of upright fireboxes of donkey or similar boilers, 10 per cent. should be deducted from the constant given above, applicable to the respective classes of work.

furnaces, but the constants should be reduced as 30 to 47.

In all cases the internal steam pipes should be so fitted that the steam in flowing to them will pass over all the plates which have steam in contact with them, and are exposed to the impact of heat or flame.

Superheaters should, as regards survey, be deemed to be the *most important parts* of the boilers, and must be inspected *inside and outside*; those that cannot be entered on account of their size and arrangement must have a sufficient number of doors through which a thorough inspection of the whole of the interior can be made.

Special attention should be paid to the survey of superheaters, as with high pressures the plates may become dangerously weak, and not give any sound to indicate their state when tested with a hammer; the plates should therefore be occasionally drilled.

Before commencing the survey, it is prudent to question the Engineer Officers as to the tendency of the boilers to flame; if flaming is a frequent occurrence, extra care must be taken in the survey, and in fixing the pressure to be allowed, as the tensile strength of the plate, when heated, is often reduced to a few tons per square inch. Drain pipes must in all cases be fitted to superheaters in which a collection of water in the bottom is possible.

Safety-valve for superheaters.

Superheaters that can be shut off from the main boilers should be fitted with a parliamentary safety-valve of sufficient size, but the least size passed without special written authority should be three inches diameter.

The flat ends, &c., of all boilers, as far as the steam space extends, and the ends of superheaters, should be fitted with shield, or baffle plates, where exposed to the hot gases in the uptake.

90. As the uptakes of haystack boilers and others of similar type are especially liable to injury from overheating unless careful precautions are taken while steam is being raised, the Surveyor should in all cases endeavour to persuade makers and owners of such boilers to make the strength of the uptakes considerably in excess of that required for ordinary superheaters subject to external pressure. **Haystack boilers.**

The employment of Bowling rings is beneficial by adding to the strength as well as allowing for expansion, but if there is a difficulty in getting these fitted, hoops riveted to the uptake, although not so desirable as Bowling rings, may be employed to increase the resistance of the tubes against collapse. The use of Bowling rings with a moderate thickness of plate is better than very thick plating. This applies to the uptakes of all boilers of this type, including ordinary vertical donkey boilers. When flaming coal is used extra care is required and extra strength absolutely necessary.

91. Evaporators, generators, feed make-ups, &c., where the evaporation of water under pressure is an essential feature, should be regarded as steam boilers, whether the evaporation is effected by heat from coal gas, from steam, or from any other source, and the strength, quality of material, and method of construction of such apparatus, should be in accordance with the regulations for steam boilers, and they should be examined by the Surveyor on each occasion the vessel is surveyed for passenger certificate in the same manner as other boilers on board the vessel. **Evaporators, generators, feed make-ups, &c.**

The mountings, &c., should as a general rule be similar to those required in the case of boilers on board passenger vessels.

When a reducing nozzle is fitted in the steam supply pipe the contracted orifice should not in

ordinary cases exceed that found by the following formula:—

$$\frac{A \times p}{6 \times P} = \text{area of orifice.}$$

A = combined area of safety valves fitted to the evaporator.

p = absolute pressure at which the evaporator is worked.

P = absolute pressure of entering steam.

The particulars of evaporators, their safety valves, &c., should be recorded on the declaration in the same manner as is done in the case of other auxiliary boilers.

Steel Boilers.

Maker's tests.

92. The following should guide the Board's Surveyors when the general quality of the steel has been found suitable for marine boilers.

The steel makers or boiler makers should test one or more strips or pieces cut from *each* plate for tensile strength and elongation, and stamp both results on each plate. When practicable the plates should be so stamped that the marks can be easily seen when the boiler is constructed.

Surveyor's tests of plates.

93. The Surveyor is not obliged to witness the foregoing tests, although it is very desirable that he should when his other duties will allow him to do so, but he should see that all the plates are properly marked. He should, however, select of each thickness *at least* one in four of the plates to be used in the construction of the boiler and witness the testing by pulling and bending of at least one strip or piece cut from each selected plate, but when boiler plates exceed 15 feet in length, there should be a tensile test from each end and one bend test, and when they exceed 20 feet in length and at the same time exceed 6 feet in breadth, or exceed 2½ tons in

weight, there should be a tensile test from each corner, and a bend test from each end. In the latter cases the testing of *each* plate should be witnessed by the Surveyor. The mean of the results of the tests, if the latter fall within the Board's requirement, as stated below, should be stamped on the plates. If a large number of failures take place in the 25 per cent. selected, the Surveyor should select and see tested, an additional number of plates, the number of additional tests being proportional to the number of failures. If for the plates from which the Surveyor selects the above proportion, a greater stress is wished than is allowed for iron, tests for tensile strength and elongation should be made, also a few tempering and bending tests, and those for which no reduction of thickness is asked may be tested for resistance to bending and tempering only, if preferred. In the latter case the tensile strength and elongation stamped on each plate should be reported by the Surveyor to the Board of Trade, together with the results of the bending and tempering tests. From the plates, the tests of which have been made by the steel maker, and not witnessed by the Surveyor, the Surveyor may, if he thinks it advisable, select any plates after they are in the boiler yard and require specimens to be cut off and tested. If the results are not satisfactory the whole of the plates, except those which were tested and found satisfactory by the Surveyor, may be liable to be rejected.

Tensile and bending tests.

Test strips.

94. The breadth of test strips for tensile stress should be, where practicable, 2 inches, and the elongation taken in a length of 10 inches, should be about 25 per cent., and not less than 18 per cent. when the strips are tested in the normal condition, in which condition the Board prefers the tests to be made; but if the plates are annealed, that is, heated

to a red heat in a *plate* furnace, and immediately they are at that heat taken out and placed on the mill floor to cool, the elongation should not be less than 20 per cent.. The test pieces must not be annealed after they are cut off the plates. When the plates are not taken out of the furnace immediately they are red hot, or if allowed to cool down in the furnace, or are covered with ashes or other non-conducting substance, it should be reported to the Board for their consideration and decision. The Surveyor should report to the Board whether the plates were annealed, or in the normal condition when the test pieces were cut off. The test strips must be carefully prepared and measured, and, where practicable, they should be cut from the plate by a planing or shaping machine.

The Surveyor should see that the plates for the manhole doors and for the compensating rings around the openings for the doors are tested in the usual manner.

Bending tests.

95. The bending tests for plates *not* exposed to flame should generally be made with strips in the same condition as the plates, but occasionally some tempering tests should be made. Strips cut from furnaces, combustion boxes, &c., and also those plates which will be worked in the fire, should be heated to a cherry red, then plunged into water at about 80° and kept there until of the same temperature as the water, and then bent. The bending and tempering strips should be about 2 inches broad and 10 inches long, and they should be bent until they break, or until the sides are parallel at a distance from each other of not more than three (3) times the thickness of the plates.

[illegible]sile [illegible]h, &c.

96. When full allowance over iron is wished, the tensile strength of the plates *not* exposed to flame

should be not less than 27 tons, and should not exceed 32 tons per square inch of section, and 27 tons should be the stress used in the calculations for cylindrical shells if the plates comply with all the conditions as stated herein, but for each ton the minimum tensile strength of the plate is above 27 tons, 1 ton may be added to the 27 used in the calculations, provided the Surveyor witnesses the testing of *all* the plates. The tensile strength of furnace, flanging, and combustion box plates, may range from 26 to 30 tons per square inch.

97. The following proportion of stay and rivet bars should be tested in the Surveyor's presence for tensile strength and elongation, viz.: one bar in 20 when the diameter of the bar does not exceed 1 inch; one bar in 12 when not over 1½ inches; and one bar in 8 when the diameter exceeds 1½ inches. When the number of bars of any one size in the order exceeds the number for which the Surveyor is required to make one test, but is less than double that number, he should make two tests from bars of that size.

Surveyor's test of stay and rivet bars.

The tensile strength of stay bars should be from 27 to 32 tons per square inch, with an elongation of about 25 per cent. and not less than 20 per cent. in a length of 10 inches. Solid steel screwed stays may be allowed a working stress of 9000lbs. per square inch of net section, provided the tensile strength and elongation are as stated. Steel stays which have been welded should not be passed. (This does not apply to stay tubes which are welded longitudinally.)

Tensile strength of stays, &c.

Solid steel stays for supporting dished ends, which are found to be equal to the pressure needed when considered as portions of spheres may have a

Steel stays for dished ends.

not less than $\frac{5}{16}$ inch thick, may be allowed the working pressure found by the following formula:—

$$\frac{C \times T}{D} = \text{working pressure.}$$

C = *14000.

T = thickness in inches, measured at the bottom of the corrugation or camber.

D = outside diameter in inches, measured at the bottom of the corrugations or cambers when the furnace is of the corrugated or cambered type, or over the plain parts, when it is of the ribbed and grooved description.

In the Fox furnace the pitch of the corrugations should not exceed 6 inches, and in the Morison furnace and the Deighton furnace the pitch should not exceed 8 inches. In these descriptions of furnaces the depth from top of corrugations outside to bottom of corrugations inside should not be less than 2 inches

The ribs of ribbed and grooved furnaces should not be less than $1\frac{5}{16}$ inches above the plain parts, the depths of the grooves not more than $\frac{3}{4}$ inch, and the length between the centres of the ribs not over 9 inches. In Brown's cambered furnace the thickness of metal at the centre of the ribs should be at least $\frac{3}{16}$ inch greater than the thickness at the bottom of the camber, the tops of the ribs should be curved to a radius of $1\frac{3}{8}$ inches and the grooves beneath the ribs to a radius of $\frac{45}{64}$ inch, the height of the ribs above the bottom of the camber should not be less than $1\frac{5}{8}$ inches and the pitch of the ribs should not be more than 9 inches.

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*This constant only applies to furnaces of the type named when made by the firms given in the preceding part of this paragraph. The Surveyors should continue to report full particulars of any case in which the owners or builders of a passenger steamer propose to use furnaces of any of these types if made by other makers.

Machine made furnaces of the bulb type manufactured by the Leeds Forge Company may be allowed the working pressure found by the following formula, provided they are practically true circles, that the pitch of the bulbs does not exceed 8 inches, that the depth from the top of the bulbs to the plain parts at the centre of the pitch is not less than $2\frac{1}{4}$ inches, that the plates are not less than $\frac{5}{16}$ inch thick and the plain parts between the bulbs are fairly uniform in thickness:—

$$\frac{15000 \times T}{D} = \text{working pressure.}$$

T = thickness of the plain parts between the bulbs in inches.

D = outside diameter at the middle of the plain parts between the bulbs in inches.

In each of these descriptions of furnace the plain parts at the back ends should be so made that the length, measured from the waterside of the back tube plate to the centre of the back end corrugation or rib, does not exceed 9 inches. The plain parts at the front ends should also be so made that the length of the flat, measured from the centre of the rivets by which the furnace is secured to the front end plate, does not exceed 9 inches. When the plain parts at the back ends are made conical, and the flange by which the attachment is made to the back tube plate is continuous, a length of $10\frac{1}{2}$ inches may be allowed between the waterside of the back tube plate and the centre of the first corrugation or rib. When this method of construction is adopted, the vertical section through the neck-piece should be kept as circular as is practicable, the set up at the bottom should not exceed 8 inches measured over the plates, and in no case should the vertical axis exceed the horizontal one by more than $14\frac{1}{2}$ per cent. The plates at the ends should not be unduly thinned in the flanging.

If the furnace is riveted in two or more lengths the case should be submitted for consideration.

Furnaces made up of flanged rings.

101. When horizontal furnaces of ordinary diameter are constructed of a series of rings welded longitudinally, and the ends of each ring flanged and the rings riveted together, and so forming the furnace, the working pressure is found by the following formula, provided the length in inches between the centres of the flanges of the rings is not greater than (120 T—12), and the flanging is performed at one heat by a suitable flanging machine, and also the conditions which follow the formula are complied with:—

$$\frac{9900 \times T}{3 \times D}\left(5 - \frac{l + 12}{60 \times T}\right) = \text{working pressure.}$$

T = thickness of plate in inches.
l = length between centre of flanges in inches.
D = outside diameter of furnace in inches.

The radii of the flanges on the fire side should be about 1½ inches. The depth of the flanges from the fire side should be three times the diameter of the rivet plus 1½ inches, and the thickness of the flanges should be as near the thickness of the body of the plate as practicable. The distance from the edge of the rivet holes to the edge of the flange should not be less than the diameter of the rivet, and the diameter of the rivets at least ⅜ inch greater than the thickness of the plate. The depth of the ring between the flanges should not be less than three times the diameter of the rivets, the fire edge of the rings should be at about the termination of the curve of flange, and the thickness not less than half the thickness of the furnace plate. It is very desirable that these rings should be turned.

The holes in the flanges and rings should be drilled in place if practicable, but if not drilled in place they should be drilled smaller than the size required,

and afterwards when in place rimered out until the holes are quite fair, the holes should be slightly tapered and the heads of the rivets of moderate size.

After all the welding, flanging, and heating is completed each ring should be efficiently annealed in one operation.

When the flanges of the back ends of the furnaces are not continuous, and the lower parts of the back rings are supported by substantial T-bars securely riveted to the plates, the constant use for these rings should not exceed $\frac{9}{10}$ths of that given in the formula.

102. A greater compressive stress should not be allowed on tube plates than 14000lbs., which is that used in the following formula:— Compressive stress on tube plates.

$$\frac{(D - d)\ T \times 28000}{W \times D} = \text{working pressure}$$

D = least horizontal distance between centres of tubes in inches.

d = inside diameter of ordinary tubes in inches.

T = thickness of tube plate in inches.

W = width of combustion box in inches between the tube plate and back of fire box, or distance between the combustion box tube plates when the boiler is double-ended and the box common to the furnaces at both ends.

103. When the minimum tensile strength of the shell plates is S tons and full allowance is wished the rivet section, if iron, in the longitudinal seams of cylindrical shells should, when those seams are lapped, be at least $\frac{S}{17 \cdot 5}$ times the net plate section, and if steel rivets are used their section should be at least $\frac{S}{23}$ of the net section of the plate if the tensile strength of the rivets is not less than 27 tons and not more than 32 tons per square inch. In calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the Board's rules, but in dealing with iron rivets the percentages Plate and rivet section.

found should be divided by $\frac{8}{17 \cdot 5}$, and in the case of steel rivets by $\frac{8}{23}$, the results being the percentages required. If the percentage strength of the rivets is found by calculation to be less than the calculated percentage strength of the plate, the working pressure should be calculated by both percentages. When using the percentage strength of the plate 4·5 plus the additions suitable for the method of construction as by the Board's rules for iron boilers, may be used as the nominal factor of safety, but when using the percentage strength of the rivets 4·5 may be used as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell. (See also the formulæ in Appendix A.)

Local heating to be avoided.

104. Local heating of the plates should be avoided, as many plates have failed from having been so treated.

Annealing

All plates that have been punched, flanged, or locally heated, also all stays and stay tubes which have been locally heated should be carefully annealed after being so treated.

Welding, &c.

Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

Steel tubes made by the Mannesmann process need not be objected to for use in boilers, provided the material and the tests comply in all respects with the Board's usual requirements.

In other respects the boilers should comply with the rules for iron boilers.

If the tests are to be made at the steel works the boiler makers should inform the Surveyors in their district when and where they will be made, so that a Surveyor from the nearest district to the steel works may be instructed to attend to them. The Surveyors should have due notice—two or three

days—when the plates, &c., will be ready for the test pieces to be cut from them. As soon as possible after tests are made, the results should be submitted for the Board's consideration. (Surveyors should report all cases of failures of steel plates, &c., which may come to their knowledge.)

105. If steel is proposed to be used for the heating surfaces in superheaters the particulars should be submitted to the Board of Trade for consideration, but in all cases it should be discouraged for this purpose. This applies to the unshielded uptakes of all boilers including ordinary vertical donkey boilers. **Steel for superheaters, &c.**

106. When the steel is not to be made by any of the following makers the case will receive the special consideration of the Board, and this should be specially noted by the Surveyors. **Makers of steel.**

Messrs. W. Beardmore & Co.
„ J. Brown & Co.
„ Cammel, Laird & Co.
„ The Clydebridge Steel Co.
„ D. Colville and Sons.
„ The Consett Iron Co., Ltd.
„ The Glasgow Iron and Steel Co., Ltd.
„ Guest, Keen and Nettlefolds, Ltd.
„ The Leeds Forge Co.
„ Palmer's Shipbuilding and Iron Co., Ltd.
„ The South Durham Steel and Iron Co., Ltd.
„ John Spencer and Sons. Ltd.
„ The Steel Co. of Scotland.
„ The Weardale Steel, Coal and Coke Co., Ltd.
„ Stewarts and Lloyds, Ltd.

For plates, angles, stay and rivet bars.

Messrs.	J. Dunlop and Co.	For plates.
„	Baldwins, Ltd.	For stay and rivet bars.
„	The Lanarkshire Steel Co., Ltd.	
„	Dorman, Long & Co., Ltd.	For stay and rivet bars, angles and flats.
„	The Wigan Coal and Iron Co., Ltd.	

Boiler tracings, &c.

107. Difficulty has been experienced with regard to the survey of steel boilers owing to the fact that some makers were not aware, at the time the construction of the boilers was commenced, that a Board of Trade certificate would be necessary, and the makers have therefore omitted to submit tracings until the boilers have been nearly completed. Tracings of boilers may therefore be received for examination upon payment of the usual fee, and the Surveyors may proceed as far as witnessing the hydraulic test and making the subsequent internal examination before any further instalment of the survey fee is paid. Engineers and boiler makers should be advised of this arrangement, and should be informed that the fee for the survey of one boiler is £2, and for each additional boiler made to the same design and for the same vessel a further fee of £1.

Boilers.—General.

Donkey boilers.

108. Donkey boilers that are in any way attached to or connected with the main boilers, or with the machinery used for propelling the vessel, should be surveyed and have their working pressure fixed in the same way as the main boilers, and have a water and steam gauge, and all other fittings complete, and, as regards safety-valves, should comply with the same regulations as the main boilers.

109. The boilers of steam launches forming part of the statutory boat capacity of passenger steamers should as regards construction, strength, material, safety-valves, and other fittings comply with the same regulations as the main boilers. Launch boilers.

Stop-valves.

110. No boiler or steam chamber should be so constructed, fitted, or arranged that the escape of steam from it through the safety-valves required by the Act of Parliament can be wholly, or partially, intercepted by the action of another valve.

A stop-valve should always be fitted between the boiler and the steam pipe, and, where two or more boilers are connected with a steam receiver or superheater, between each boiler and the superheater or steam receiver. The necks of stop-valves should be as short as practicable.

111. Each boiler should be fitted with a glass water-gauge, at least three test cocks, and a steam gauge. Boilers that are fired from both ends, and those of unusual width, should have a glass water-gauge and three test cocks at each end or side, as the case may be. An additional glass water-gauge may, however, be substituted for three test cocks. When a steamer has more than one boiler each boiler should be treated as a separate one, and have all the requisite fittings. Water-gauges test cocks, steam-gauges, &c.

When the water-gauge cocks are not attached directly to the shell of the boiler, but to a stand pipe or column, cocks should as a general rule be fitted between the boiler and the stand pipes, &c., and may be placed either on the boiler or at the stand pipe. Such cocks need not, however, be insisted on in cases where the columns, stand pipes, &c., are of moderate length and of suitable strength, provided that the diameter of the bore at any part is not less than 3 inches. Valves should not be passed between the boiler and the stand pipe.

If the column, stand pipes, &c., are of less diameter than 3 inches, and the pipes are bolted to the boiler without the intervention of cocks, the arrangement need not be objected to, if otherwise satisfactory, providing there is no difficulty in keeping the passage at the ends clear, and ascertaining that they are so. To do this it will be necessary that the passage in the part of the column between the top and bottom gauge-glass cocks be cut off or closed, which may be done permanently, or by the interposition of a cock at that part. The latter is a convenient and desirable arrangement even when cocks are fitted on the boiler.

In the case of high pressure boilers, it is desirable that the cocks in connection with the water-gauges should be fitted with handles which can be expeditiously manipulated from a convenient position.

It is desirable in all cases that test cocks should be fitted directly to the skin of the boiler, and when the water-gauge is attached to a column, the opening through which is stopped or can be cut off, the test cocks *must* be fitted directly to the skin of the boiler.

The Surveyors should satisfy themselves by actual examination whether the glass water-gauges of the boilers of the vessels they survey are clear, and also whether they are fitted with automatic valves or fittings, as the existence of such fittings cannot always be ascertained by external examination. In all cases where automatic gauges are fitted, full particulars thereof should be submitted for consideration and approval before the gauges are passed.

No new arrangement to be sanctioned until plans have been submitted to Board of Trade.

112. Surveyors are to be most careful not to give any official sanction to any new or unusual arrangement or construction of marine steam boilers, without first obtaining the permission of the Board, nor

should they give any written approval of any invention, or arrangement, unless by direction of the Board of Trade; and whenever they know that any new arrangement of boiler or other apparatus is to be fitted to a vessel that is intended to have a passenger certificate, they should as soon as possible, with a view to prevent subsequent delays and question, obtain plans of it and submit them for the consideration of the Board of Trade.

When any deviation from an approved plan is made, full particulars thereof should be submitted for the Board's consideration, and when any deviation is sanctioned it is only for that particular case, unless otherwise stated.

Surveyors should in all cases record on their declarations whether the boilers are made of iron or steel, and if made partly of steel and partly of iron they should specify for what parts either metal is used.

Inside of boilers to be examined every survey.

113. At every survey of a passenger steam ship the Board of Trade desire and expect the Surveyor to go inside the boilers, and make a thorough examination of them. Drilling of furnace and the lower parts of combustion box plates, shell plates, &c., should of course be done when considered necessary in order to ascertain their actual thickness.

Declarations must not be granted for boilers which the Surveyor is unable to enter in consequence of the manholes not being large enough, or being improperly placed, until sufficient means of access are provided.

When stays alone prevent the Surveyor getting into a boiler, such stays should be removed, but he must see them properly replaced before granting his declaration.

If any part of a main boiler near the uptake, fireboxes, or furnaces is so constructed that the Sur-

veyor cannot examine it, he should withhold the declaration, and report the case to the Senior Engineer Surveyor, who will, if necessary, refer it for instructions.

Boilers too small for the Surveyor to enter.

When *boilers* are not large enough for the Surveyors to enter, they should be examined as far as possible, and tested by hydraulic pressure at every annual survey at least. The hydraulic test may also be applied at every six monthly survey if the Surveyor considers it necessary .

If for special reasons, and for special reasons only, the Surveyor cannot go inside a donkey boiler or other small boiler, he must distinctly state, on the face of his declaration, his reason for not being able to do so.

Surveyors not to go into hot boilers.

If the boiler is too hot for the Surveyor to examine the inside efficiently, and with safety and convenience, he should decline to examine it, and refuse to grant a declaration, until he can make an efficient internal examination.

Surveyors who have to enter a boiler, which it is possible to connect with another boiler containing steam, should, before doing so, take all reasonable precautions to secure immunity from the danger of steam being turned into the boiler which he enters during the time he is inside it.

Precaution regarding broken stays.

The Surveyors are cautioned that a number of cases have recently occurred in which many screwed steel stays in the water spaces of steel boilers have broken squarely across in line with the water side of the outer plate in the ordinary course of working. It is sometimes difficult to detect such defects, although they may be so serious as to render the boiler in which they occur quite unfit for the pressure required, and for which it would otherwise be sufficient. Great vigilance is therefore necessary in the examination of such stays, and the Surveyors

will find the frequent application of the hydraulic test a valuable aid to the discovery of broken ones. The surfaces supported by these stays should be carefully gauged, before, during, and after the hydraulic test, as undue deflection will no doubt occur in the vicinity of any stays that may be broken. It is also desirable that the outer ends of all such stays should have a small hole drilled axially, of sufficient depth to extend into the body of the stay at least ½ inch beyond the plate into which the stay is screwed. If that is done, the breaking of a stay in the above manner will result in leakage, and give warning to those in charge. Some makers reduce the diameter of the body of such stays somewhat below that of the ends where screwed into the plates; that practice is a commendable one, and should be encouraged by the Surveyors.

114. Before requiring a boiler to be tested by hydraulic pressure, the Surveyor should examine it as far as possible, take the necessary measurements, and calculate the working pressure for it by the Board of Trade rules. Strength of boilers to be ascertained and working pressure fixed by calculation.

This instruction applies to superheaters and steam chests, as well as to boilers, evaporators, &c.

115. Surveyors should see all new boilers, and boilers that have been taken out of the ship for a thorough repair, tested by hydraulic pressure to double the working pressure that will be allowed. The test should be made previous to the boilers being placed in the vessel, and before they are lagged. Hydraulic test.

The hydraulic test should not be witnessed by the Surveyors in any case where the Board's regulations as to strength, material, method of construction, treatment, &c., are not complied with unless they have previously submited the details of the particu-

lar case for the consideration of the Board and obtained authority to witness the test.

The hydraulic test should not be applied until the boiler has been examined in accordance with Clause 113, and until the strength has been calculated from the necessary measurements taken from the boiler itself as per Clause 114.

When the boilers are in the vessel the Surveyor may, at any time he thinks it necessary, before he gives a declaration, require them to be tested by hydraulic pressure to satisfy himself as to the sufficiency or efficiency of any doubtful part, or of any part not easy of access for inspection.

The *full* hydraulic test should be applied to the boilers of all steamers that have not previously had a passenger certificate, before a declaration is granted for them.

The *full* hydraulic test should be applied at each annual survey to boilers which are too small for the Surveyor to enter or satisfactorily examine internally. After a boiler has been subjected to the hydraulic test the Surveyor should inspect it, as far as possible, both externally and internally.

If, while a boiler is being tested, there are any visible or audible indications of its being defective the Surveyor should at once advise those conducting the test to relieve the boiler of pressure, and take steps to ascertain the nature and extent of the defect. The Surveyor's primary duty at a test is, however, to note the results and satisfy himself that it is properly made, the conduct of the test being left to the representatives of those who own the boiler. When a test is unsatisfactory the defects should be made good and the boiler re-tested.

No test should be considered good in which the boiler has not borne satisfactorily the intended test pressure for at least ten consecutive minutes.

The amount of the test pressure and the date on which the test was last applied should in all cases be inserted in the Surveyor's declaration, and recorded in the office boiler book.

116. Surveyors should pay particular attention to the examination and testing of steam pipes. Examinat and testi steam pi

All new copper steam pipes should be tested by hydraulic pressure to not less than twice and not more than two and one-half times the working pressure. The higher test should be that usually employed. When, however, special considerations arise, the case should be fully submitted and instructions obtained before the Surveyor proceeds with the hydraulic test.

Wrought-iron lap-welded steam pipes should be tested by hydraulic pressure, when new, to at least three times the working pressure, but a higher test pressure need not be objected to provided it does not exceed four times the pressure found by the rule in Clause 118.

As regards old pipes the Surveyor may at any time he thinks it necessary, before he gives a declaration, require them to be tested by hydraulic pressure to satisfy himself as to any doubtful part, but they should be tested periodically, with the lagging removed for examination, to not less than double the working pressure. A record of the test should be kept in the office boiler book. For explanatory and supplementary instructions *see* Appendix B, page 323.

There should be efficient means provided for draining all steam pipes. Boiler stop-valves cannot be regarded as suitable for that purpose. All drain cocks or valves should be accessible and so placed as to render it practicable to drain the water from any portion of the steam pipes or chests in connection therewith. Drain pipes should be fitted to Draining steam pip

drain cocks or valves when the latter are in such a position that the water or steam discharged therefrom would be likely to cause personal injury. It is desirable that the drains should be automatic in their action.

Copper pipes. 117. The working pressure of well-made copper pipes when the joints are brazed is found by the following formula:—

$$\frac{6000 \times (T - \frac{1}{16})}{D} = \text{working pressure.}$$

T = thickness in inches.

D = inside diameter in inches.

When the pipes are solid drawn and not over 10 inches diameter substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

In any case where the Board of Trade authorise a trial to be made of electro deposited pipes, the Surveyors should continue, as at present, to see that the conditions on which such trial is authorised are fulfilled.

Wrought-iron pipes. 118. The internal pressure on wrought-iron pipes made of good material and lap-welded may be determined by the following formula, provided that the minimum thickness is not less than $\frac{1}{4}$ inch, and the workmanship, hydraulic test, &c., satisfactory:—

$$\frac{6000 \times T}{D} = \text{working pressure.}$$

T = thickness in inches.

D = inside diameter in inches

Feed pipes. 119. Feed pipes, feed heaters, filters or other vessels through which the feed water passes on its way from the pumps to the boilers, should be made sufficient for a pressure 20 per cent. in excess of the boiler pressure.

Expansion joints. 120. In all cases in which a socket expansion joint is fitted to a bent steam pipe, the Surveyor should require a fixed gland and bolts or other efficient

means to be provided to prevent the end of the pipe being forced out of the socket. This regulation should be complied with in all cases of bent pipes fitted with socket expansion joints, and it is also desirable that fixed glands and bolts should be fitted to the expansion joints of *straight* steam pipes, as cases have occurred, particularly with small straight pipes, in which the ends have been forced out of the sockets.

Cast iron in boilers, steam-pipes, stand-pipes, cocks, &c.

121. In all boilers in which the Surveyors find that cast iron is employed in such a manner as to be subjected to the pressure of steam or water, they should report the circumstances to the Board of Trade. Cast-iron standpipes or cocks intended for the passage through them of hot brine should not be passed. Surveyors should also discourage the use of cast-iron chocks and saddles for boilers, and particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

Pressure fixed by one Surveyor not to be increased by another.

122. When the boilers of a vessel are surveyed in the works by one Surveyor and the survey of the vessel is completed and the declaration granted by another, the Surveyor who has surveyed the boilers should inform the Surveyor who is to grant the declaration what, in his opinion, the working pressure should be; he should also certify in the usual manner as to the efficiency and sufficiency of the parts he has surveyed. The Surveyor who grants the declaration must also examine the boilers after they are fitted in the vessel, and in the absence of special instructions from the Board of Trade he will be held responsible for the pressure allowed. He should, however, neither reduce nor increase the pressure named by the Surveyor who surveyed the boilers in the works without consulting that officer, and if any difference of opinion then exists the matter should be referred to the Board of Trade.

A pressure once allowed on the boiler of a passenger steam ship should not, *under any circumstances*, be increased unless the Surveyor has previously referred the matter to the Board of Trade. In cases where a Surveyor is of opinion that an increased pressure may with safety be allowed, he should communicate with the Surveyor who last surveyed the vessel; and, if, on learning the reasons why the existing pressure was formerly allowed, the Surveyor is still of opinion that it may be increased, he should communicate all the facts of the case to the Board of Trade; but, as above stated, the pressure should not in any case be increased until the question has been decided by them.

Safety-Valves.

Provisions as regards safety-valves.

123. The locked-up valves, *i.e.*, those out of the control of the engineer when steam is up, should have an area not less, and a pressure not greater, than those which are not locked up if any such valves are fitted.

Cases have come under the notice of the Board of Trade in which steam ships have been surveyed, and passed by the Surveyors with pipes between the boilers and the safety-valve chests. Such arrangement is not in accordance with the Act, which distinctly provides that the safety-valves shall be upon the boilers.

The Surveyors are instructed that in all *new boilers*, and whenever *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible and be cast in one with the chest.

The Surveyors should note that it is not intended by this instruction that vessels with old boilers which have been previously passed with such an

arrangement should be detained for the alterations to be carried out.

Of course, in any case in which a Surveyor is of opinion that it is positively dangerous to have a length of pipe between the boilers and the safety-valve chest, it is his duty at once to insist on the requisite alterations being made before granting a declaration.

If any person place an undue weight on the safety-valve of *any* steam ship, or in the case of steam ships surveyed under the Act, increase such weight beyond the limits fixed by the Engineer Surveyor he shall, in addition to any other liability he may incur by so doing, be liable for each offence to a fine not exceeding one hundred pounds.

Area of safety-valves.

124. When natural draught is used the area per square foot of fire-grate surface of the locked-up safety-valves should not be less than that given in the following table opposite the boiler pressure intended, but in no case should the valves be less than 2 inches in diameter. This applies to new vessels and to vessels which have not previously received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the table, it will be necessary to fit them either with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits.

When forced draught is used, the area of the safety-valves should not be less than that found by the following formula:—

$$\frac{A \times \left\{\begin{array}{l}\text{Estimated consumption of}\\ \text{coal per square foot of}\\ \text{grate in lbs. per hour}\end{array}\right.}{20} = \begin{array}{c}\text{area of valves}\\ \text{required.}\end{array}$$

A = area of valves as found from the table.

When the pressure exceeds 180lbs. per square inch the accumulation of pressure at the steam test will probably be exceptionally high, unless the area of the branch leading from the valve chest is in excess of the area of the valves and the area of the main waste steam pipe is correspondingly in excess of the gross area of the valves.

When ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

In the case of vessels that have not previously had a passenger certificate, if there is only one safety-valve on any boiler, the Surveyor should not grant a declaration without first referring the case to the Board for special instructions.

SAFETY-VALVE AREAS.

(Natural Draught.)

Boiler Pressure.	Area of Valve per square foot of Fire-grate.	Boiler Pressure.	Area of Valve per square foot of Fire-grate.	Boiler Pressure.	Area of Valve per square foot of Fire-grate.
15	1·250	36	·735	57	·520
16	1·209	37	·721	58	·513
17	1·171	38	·707	59	·506
18	1·136	39	·694	60	·500
19	1·102	40	·681	61	·493
20	1·071	41	·669	62	·487
21	1·041	42	·657	63	·480
22	1·013	43	·646	64	·474
23	986	44	·635	65	·468
24	·961	45	·625	66	·462
25	·937	46	·614	67	·457
26	·914	47	·604	68	·451
27	·892	48	·595	69	·446
28	·872	49	·585	70	·441
29	·852	50	·576	71	·436
30	·833	51	·568	72	·431
31	·815	52	·559	73	·426
32	·797	53	·551	74	·421
33	·781	54	·543	75	·416
34	.765	55	·535	76	·412
35	.750	56	·528	77	·407

SAFETY-VALVE AREAS—*continued.*

Boiler Pressure.	Area of Valve per square foot of Fire-grate.	Boiler Pressure.	Area of Valve per square foot of Fire-grate.	Boiler Pressure.	Area of Valve per square foot of Fire-grate.
78	·403	126	·265	174	·198
79	·398	127	·264	175	·197
80	·394	128	·262	176	·196
81	·390	129	·260	177	·195
82	·386	130	·258	178	·194
83	·382	131	·256	179	·193
84	·378	132	·255	180	·192
85	·375	133	·253	181	·191
86	·371	134	·251	182	190
87	·367	135	·250	183	·189
88	·364	136	·248	184	·188
89	·360	137	·246	185	·187
90	·357	138	·245	186	·186
91	·353	139	·243	187	·185
92	·350	140	·241	188	·184
93	·347	141	·240	189	·183
94	·344	142	·238	190	·182
95	·340	143	·237	191	·181
96	·337	144	·235	192	·181
97	·334	145	·234	193	·180
98	·331	146	·232	194	·179
99	·328	147	·231	195	·178
100	·326	148	·230	196	·177
101	·323	149	·228	197	·176
102	·320	150	·227	198	·176
103	·317	151	·225	199	·175
104	·315	152	·224	200	·174
105	·312	153	·223	201	·173
106	·309	154	·221	202	·173
107	·307	155	·220	203	·172
108	·304	156	·219	204	·171
109	·302	157	·218	205	·170
110	·300	158	·216	206	·169
111	·297	159	·215	207	·169
112	·295	160	·214	208	·168
113	·292	161	·213	209	·167
114	·290	162	·211	210	·166
115	·288	163	·210	211	·166
116	·286	164	·209	212	·165
117	·284	165	·208	213	·164
118	·281	166	·207	214	·164
119	·279	167	·206	215	·163
120	·277	168	·204	216	·162
121	·275	169	·203	217	·161
122	·273	170	·202	218	·161
123	·271	171	·201	219	·160
124	·269	172	·200	220	·159
125	·267	173	·199		

Examination of safety-valves.

125. The Surveyor in his examination of the machinery and boilers is particularly to direct his attention to the safety-valves, and whenever he considers it necessary, he is to satisfy himself as to the pressure on the boiler by actual trial.

The Surveyor is to fix the limits of the weight to be placed on the safety-valves, and the responsibility of issuing a declaration before he is fully satisfied on the point is very grave. The law places on the Surveyor the responsibility of "declaring" that the boilers are in his judgment sufficient with the weights he states.

The Surveyor is to examine the whole of the valves, weights, and springs at every survey.

The responsibility of seeing to the efficiency of the mode by which the valves are fitted, so as to be out of the control of the engineer when steam is up, rests with the Surveyor, who should see that the method adopted is efficient and approved by the Board of Trade.

The safety-valves should be fitted with lifting-gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting-gear should in all cases be arranged so that it can be worked by hand either from the engine room or stoke-hole.

Care should be taken that the safety-valves have a lift equal to one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste-steam pipe, have each an area not less than the area required by Clause 124, and the area of the main waste steam pipe should not be less than the combined area of the branch pipes. Each valve box should have a drain pipe fitted at its lowest part.

In the case of lever-valves, if the holes in the lever are not bushed with brass, the pins must be

of brass; iron and iron working together must not be passed.

Too much care cannot be devoted to seeing that there is proper lift, also that free means of escape for the waste steam are provided, as it is obvious that unless the means for escape of the waste steam are ample, the effect is the same as reducing the area of the valves or putting an extra load upon them. The valve seats should be secured by studs and nuts.

The Surveyors are, as far as in their power, to make the opinion of the Board on these points generally known to the owners of passenger steamers.

Surveyor to see valves weighted.

126. When the Surveyor has determined the working pressure he is to see the safety-valves weighted accordingly, and the weights or springs fixed in such a manner as to preclude the possibility of their shifting or in any way increasing the pressure. The limit of the weight on the valves is to be inserted in the declaration, and should it at any time come to a Surveyor's knowledge that the weights have been shifted, or the loading of the valves otherwise altered, or that the valves have been in any way interfered with, so as to increase the pressure, without the sanction of the Board of Trade, he is at once to report the facts to the Board of Trade.

Spring safety-valves.

127. If the following conditions are complied with the Surveyor need raise no question as to the substitution of spring-loaded valves for dead-weighted valves:—

(1.) That at least two valves are fitted to each boiler.

(2.) That the valves are of the size, as by Clause 124.

(3.) That the springs and valves are so cased in that they cannot be tampered with.

(4.) That provision is made to prevent the valves flying off in case of the springs breaking.

(5.) That the requisite safety-valve area is cased in and locked up in the usual manner of the Government valves.

(6.) That screw lifting gear is provided to ease all the valves, as by Clause 125.

(7.) That the size of the steel of which the springs are made is in accordance with that found by the following formula:—

$$\sqrt[3]{\frac{s \times D}{c}} = d$$

s = the load on the spring in lbs.

D = the diameter of the spring (from centre to centre of wire) in inches.

d = the diameter, or side of square, of the wire in inches.

c = 8000 for round steel.

c = 11000 for square steel.

(8.) That the springs are protected from the steam and impurities issuing from the valves.

(9.) That when valves are loaded by direct springs the compressing screws abut against metal stops or washers, when the loads sanctioned by the Surveyor are on the valves.

(10.) That the springs have a sufficient number of coils to allow a compression under the working load of at least one quarter the diameter of the valve.

The size of steel of springs of safety valves should not as a rule be less than $\frac{1}{4}$ inch.

Spring-loaded valves to be tested under steam.

128. In no case is the Surveyor to give a declaration for spring-loaded valves, unless he has examined them and is acquainted with the details of their construction, and unless he has tried them under full steam, and full firing, for at least 20 minutes with the feed-water shut off and stop-valve closed, and is fully satisfied with the result of the test. In special cases, when the valves are of novel design, or where

forced draught is used, the results of the test under full steam should be reported, but if the Surveyor is fully satisfied with them he need not delay the granting of the declaration for the vessel pending the approval of the Board. If, however, the accumulation of pressure exceeds 10 per cent. of the loaded pressure, he should withhold his declaration and report the case to the Board of Trade, accompanied by a sketch, if necessary, and stating the strength pressure and the working pressure of the boilers.

Plans of new designs or of alterations in details to be submited.

129. In the case of safety-valves, of which the principle and details have already been passed by the Board of Trade, the Surveyor need not require plans to be submitted so long as the details are unaltered, of which he must fully satisfy himself; but in any new arrangement of valves, or in any case in which any detail of approved valves is altered, he should, before assuming the responsibility of passing them, report particulars, with a drawing to scale, to the Board of Trade. He can make this drawing himself from the actual parts of the valves fitted, but in order to save time, and to facilitate the survey, the owners or makers thereof may prefer to send in tracings of their own, before the valves are placed on the boiler. If they do this the survey can be more readily made, and delay and expense may be saved to owners, as the Surveyor will not then have to spend his time, and delay the ship, in preparing drawings and comparing them with the valves.

The tracings of new safety-valve designs should, where practicable, be transmitted to the Board of Trade for consideration before the construction of the valves is commenced.

The Surveyors should arrange with manufacturers to supply the designs of safety-valves which they intend to make. An easy method of facilitating this

matter is for the manufacturer to leave in the local Surveyor's office an approved plan or plans of his valve or valves, and then afterwards to inform the Surveyors that the valves fitted are according to drawing A, B, or C, as the case may be. By this means when once a design has been agreed upon, and is adhered to, all subsequent questions and delays will be prevented.

List of designs approved.

130. The following is a List of Names of Spring Safety-valve Makers whose Standard Designs have been approved by the Board of Trade.*

Names of Firms whose Standard Designs have been approved.	Address.	Diam'ters of Valves included in Standard Designs.
		Inches.
Adams, Thomas	Manchester	2 to 6
Alley & MacLellan	Glasgow	2 to 6
Allsup & Sons	Preston	3 to 4½
Amos & Smith	Hull	2 to 5
Bailey & Leetham	Hull	3, 3¼ and 4½
Bailey, W. H. & Co. ..	Manchester	2 to 6
Blair & Co.	Stockton-on-Tees ..	3 to 4 and 6¼
Bow, McLachlan & Co. ..	Paisley	3 to 4
Central Marine Engine Works	West Hartlepool ..	3 to 4
Clarke, E. & Co.	Brimscombe, nr. Stroud	2
Clark, Geo.	Sunderland	3 to 5¾
Clarke, Chapman & Gurney	Gateshead-on-Tyne ..	3
Cochran & Co.	Annan	2 to 3
Cockburn, D.	Glasgow	2 to 6½
Cockburn, George & Co. ..	Glasgow	2 to 6½
Coe, W. J.	Liverpool	3 to 6⅞
Cox & Co.	Falmouth	2 to 3¾
Dansey	London	2
Davis, G.	Abingdon	2
Day, Summers & Co. ..	Southampton	2 to 5½
Dickinson, John, & Sons ..	Sunderland	3 to 3½
Dixon, R., & Co.	Middlesbrough ..	2½ to 3
Earle's Shipbuilding Coy...	Hull	3 to 6
Empire Company	Manchester	2 to 6
Fletcher, John	Ashton-under-Lyne ..	3 to 4
Fraser, A. B. & Co... ..	Liverpool	3 to 5½

*Note.—Any departure from the designs as approved should be reported to the Board for consideration.

Names of Firms whose Standard Designs have been approved.	Address.	Diam'ters of Valves included in Standard Designs.
		Inches.
Gourlay Brothers & Co. ..	Dundee	2 to 2½ and 3¼ to 5 $\frac{15}{16}$
Grant, J. A. & Co.	Glasgow	2 to 5
Hall, Russell & Co.	Aberdeen	3 to 4¼
Harding, Cocks & Co. ..	London	3 to 5
Harvey & Co., Ltd.	Hayle	3 to 4½
Hawthorn, R. & W., Leslie & Co.	Newcastle-on-Tyne ..	4
Henderson, D. & W. ..	Glasgow	3 to 6½
Hepple & Co.	North Shields ..	4½
Holmes, C. D. & Co. ..	Hull	4½
Hopkinson, J. & Co. ..	Huddersfield	2 to 6½
Hunt, Mitton & Co. ..	Birmingham	2 to 6
Lobnitz & Co.	Renfrew	4
London and Glasgow Engineering &c. Co.	Glasgow	3 to 6
Noakes, T. & Sons	London	2
Palmer's Shipbuilding Co.	Jarrow-on-Tyne ..	3 to 5½
Pattison, G. A.	Liverpool	3 to 6½
Paul, M. & Co.	Dumbarton	2
Plenty & Sons	Newbury	2 and 3
Pollock & Macnab	Manchester	2 and 3 to 6½
Pollock, Macnab & Highgate	Glasgow	2 and 3 to 6½
Rennoldson, J. P.	South Shields	3 and 4
Richardsons, Westgarth & Co.	Hartlepool	3¾ to 5¾
Richardsons, Westgarth & Co.	Middlesbrough ..	3 to 3½
Roger, Robert & Co. ..	Stockton-on-Tees ..	2 to 3¼
Royal Mail Steam Packet Co.	Southampton	3 to 5½
Schaffer & Budenberg ..	London	2 to 5⅛
Scott & Co.	Greenock	2⅛ to 4½
Semple, R. & Co.	Bellshill, Lanarkshire	2 to 3¾
Shepherd & Co.	Liverpool	2 to 6
Simpson, Strickland & Co.	Dartmouth	2
Sisson, W. & Co.	Gloucester	2 to 3¼
Stephenson, Robert & Co. ..	Newcastle-on-Tyne ..	3 to 6
Stevenson, J. C. & Co. ..	Preston	3
Taylor, James & Co. ..	Birkenhead	3
Thompson, J. L. & Sons ..	Sunderland	2¾ to 3
Turnbull, Alexr. & Co. ..	Glasgow	2 to 6½
Tyne General Ferry Co. ..	Newcastle-on-Tyne ..	3⅝
Wallsend Slipway Co. ..	Wallsend-on-Tyne ..	4 and 4¼
Watson, H. & Sons ..	Newcastle-on-Tyne ..	2¾ and 3
Willoughby Bros.	Plymouth	2 to 3½
Worth, Mackenzie & Co. ..	Stockton-on-Tees ..	2¼ to 3

Owners, masters, and engineers to see that safety-valves are kept in proper order.

131. It is clearly the duty of the masters and engineers of vessels to see, in the intervals between the surveys, that the locked-up safety-valves, as well as the other safety-valves and the rest of the machinery, are in proper working order. There is no provision in the Merchant Shipping Act, 1894, exempting the owner of any vessel, on the ground that she has been surveyed by the Board of Trade Surveyors, from any liability, civil or criminal, to which he would otherwise be subject. The Act of Parliament requires the Government safety-valves to be out of the control of the engineer when the steam is up; this enactment, far from implying that he is not to have access to them, and to see to their working, at proper intervals when the vessel is in port, rather implies the contrary; and the master should take care that the engineer has access to them for that purpose. Substantial locks that cannot be easily tampered with, and as far as possible weather-proof, should be used for locking up the safety-valve boxes.

Hydraulic tests and tests for accumulation of pressure to be made with Board of Trade gauges.

132. When witnessing hydraulic tests of boilers, &c., and when witnessing safety-valve tests for accumulation of pressure, the Surveyors are to use the pressure gauges supplied by the Board of Trade for the purpose. No steam gauge should be used without having a syphon filled with water between it and the boiler.

Boilers and Machinery, &c.

What machinery is to be surveyed.

133. The machinery to be surveyed comprises the engines and boilers used for propelling the vessel, and all the machinery connected therewith. A donkey boiler connected with a donkey engine that is used for pumping water into the main boiler, is considered to be connected to the machinery and boilers used for propelling the ship. But boilers and

machinery used for loading or unloading the vessel, and used exclusively for purposes unconnected with the motive power of the vessel, do not form a part of the machinery required to be surveyed by the Merchant Shipping Act, 1894.

134. Surveyors should, before granting declarations for any period exceeding six months, be careful when examining the machinery: to have (1) the upper brasses of all bearings taken off; (2) the shaft turned round and carefully examined; (3) the cylinder covers and junk rings lifted for examination of the cylinders and pistons; (4) the slide covers or bonnets removed for examination of the slides, and, if necessary, the slides taken out; (5) the air and circulating pump covers lifted for examination of the pump buckets and valves; (6) the covers of all feed and bilge pumps removed for the examination of the valves; (7) all discharge-valves and sea-cocks taken apart for examination; (8) the propeller shaft withdrawn when necessary, and it and the screw examined (in all cases in which the shaft is withdrawn, it is to be seen again after the screw is secured in its place); (9) the bridges and fire-bars removed to permit of a thorough examination of the boilers and furnaces; (10) all cocks and valves on the boilers taken apart and examined. When to be taken to pieces for examination.

When a Surveyor has seen the machinery of a new steamer in the shop before it was put together, and is satisfied with it, it will not be necessary to have it again opened up on board the ship on the first survey for examination, either by himself or another Surveyor, unless in the Surveyor's judgment it is necessary after the vessel has been tried under steam.

It will rest with owners to determine whether or not they will apply for a certificate for any period exceeding six months; but if they do so determine,

then the above instructions are to be complied with before a declaration is granted.

The above instruction are also to apply at least once a year to the survey of all passenger steamers, whatever may be the period for which the declarations are given.

After the examination of the parts has been made as above, the machinery must be tried, under steam at full pressure.

Examination to be carefully made.

In the examination of the machinery and boilers the Surveyor should in no case give a declaration without thoroughly satisfying himself that both the boilers and the machinery are sufficient for the service intended, and in good condition.

Surveys of Emigrant Steamers

135. In cases in which emigrant steamers are surveyed at the commencement of every voyage, and in which they are absent for periods less than six months at a time, the above instructions do not apply to every survey. The usual examination each voyage is sufficient. One complete survey at least, as indicated above, should, however, be made each year.

Feed check valves and cocks.

Separate feed arrangements.

136. Each boiler of a passenger vessel, whether old or new, should be fitted with suitable check valves between it and the feed pipes, and the boilers of all *new* passenger vessels, passenger tugs, and other small passenger vessels should be fitted with separate feeding arrangements in addition to, but unconnected with, the main feed pipes and valves. It is desirable that the main feed check valve chest on each boiler be separate and distinct from that of the auxiliary feed, and that a stop cock or stop valve be fitted in each chest or between each chest and the boiler, so that the latter may be shut off, and either of the check valves examined while the other feed is at work. In very small vessels an efficient hand-pump, instead of the donkey pump,

may be passed if the Surveyor has satisfied himself as to its efficiency when steam is up, and provided there are separate feed pipes and valves, as directed above. This is to apply also in the case of old vessels of the above description, when being fitted with new boilers.

The Surveyor should discourage the practice of using the same pump for the bilges and for feeding boilers.

137. All inlets or outlets in the bottom or side of a vessel near to, at, or below the deep-load water line other than the outlets of watercloset, soil, scupper, lavatory and urinal pipes should have cocks or valves fitted between the pipes and the ship's side or bottom; such cocks or valves should be attached to the skin of the ship, and be so aranged that they can be easily and expeditiously opened or closed at any time; and the cocks, valves, and the whole length of the pipes should be accessible at all times. Cocks, valves, and pipes, communicating with ship's side.

Cocks or valves standing exceptional distances from the ship's plating, that is where the necks are longer than is necessary for making the joint, should not be passed without the sanction of the Board of Trade, and one condition of their being passed is that they should be made of gun-metal, and well bracketed.

Although in the case of vessels which have already been granted passenger certificates it is very desirable that the cocks and valves above referred to should be so arranged that they can be opened and closed expeditiously by hand, and that they and the whole length of the pipes referred to should be accessible at all times, still in vessels in which there is a difficulty in attaining this object, a strict compliance with the above may be dispensed with if, after a consultation, the Principal Officer, the Senior

Engineer Surveyor and the Surveyor are satisfied that the difficulty is serious, and that the arrangements existing are on the whole safe.

The Principal Officer should in such cases always send a special report to the Board of Trade, with the Surveyor's report and statement of the arrangements on board, and of the reasons for authorising a departure from the ordinary conditions.

Guards on blow-off cocks.

138. With a view to the prevention of accidents to boilers through the blow-off cocks being left open after the boiler is run up, and to prevent water getting accidentally or intentionally into the ship by cocks being left open, all blow-off cocks and sea connections below the plates or out of sight, should be fitted with a guard over the plug, with a featherway in the same, and a key on the spanner, so that the spanner cannot be removed unless the cock is closed. The spanner when in place should extend above the platform. When cocks are in sight guards need not be fitted provided the spanners are secured to the plugs by pins. The spanners should not be shrunk on the heads of the plugs. One cock should be fitted to the boiler, and another cock on the skin of the ship or on the side of the Kingston valve.

Valves may be substituted for the blow-off cocks on the boilers, but in such cases the blow-off cock on the skin of the ship must be fitted with a spanner guard so that the spanner cannot be removed when the cock is open, and if the blow-off pipe is used for more than one boiler an intermediate switch-cock or a suitable non-return valve to each boiler should be fitted so that water cannot be blown from one boiler to another.

Circulating pipes.

When the pipes are so arranged that the water in the boilers can be circulated by means of the donkey pump, similar precautions should be observed; and

a cock fitted with a spanner and guard, the handle of which will stand above the level of the platform, should be fitted in the circulating pipes, preferably near the pump.

Non-return valves to pipes.

139. In all cases where pipes are so led or placed that water can run from the boiler or the sea into the bilge, either by accidentally or intentionally leaving a cock or valve open, they should be fitted with a non-return valve having a screw spindle not attached, by which the valve may be set down in its seat when necessary; the only exception to this is the fireman's ash cock, which must have a çock or valve on the ship's side and be above the stoke-hole plates.

Exhaust pipe of donkey engine not led through ship's side.

140. The exhaust pipe for the donkey engine should not be led through the ship's side; it should be led on deck or into the main waste steam pipe, and in all cases it should have a drain-cock on it.

Outlets of watercloset and scupper pipes.

141. In the case of the outlets of watercloset, soil, scupper, lavatory, and urinal pipes which are below the weather deck, there should be an elbow of good substantial metal other than cast-iron or lead; and the pipe connected with this elbow should, if of lead, have a sufficient bend to provide for expansion in the pipe or any movement from the working of the ship. Pipes, no matter of what material they may be constructed, should never be fitted in a direct line between the apertures in the ship's side and its connexion with the deck, or closet, or other fittings. The pipes and valves should be protected from the cargo by a substantial casing of wood or iron which need not be water-tight. Unless the watercloset and scupper pipes and their outlets are fitted in the way hereby required, or in a manner that will in the opinion of the Surveyor and of the Principal Officer be more or equally efficient and safe, the Surveyor should refuse to grant a declaration. Where closets

are fitted below the water line, as in the case of pumping closets, plans should be specially submitted for approval. For closets with their outlets above the weather deck, no special regulations are necessary.

This clause applies only to the case of sea-going steamers coming under survey for the first time.

Tests of material and fees.

142. In the case of boilers and machinery it may be impracticable without great inconvenience and delay to the shipowners to apply any satisfactory tests to the material used after the boilers or machinery are placed in the ship.

With a view to obviate such inconvenience and delay the Surveyor is authorised at the request of the manufacturer or of the person for whose intended ship the material is being manufactured, to inspect and test such material during manufacture.

Such request must be made on Form Surveys 6, or on Form Surveys 6a, or on such other form as the Board of Trade may from time to time direct, and must be accompanied with a written undertaking to pay to the Board of Trade on the delivery of the certificate, (1) such sum for travelling expenses as the Board of Trade may fix; and, (2) the sum of two guineas, or such less sum as the Board of Trade may fix as a commuted payment for every day or part of a day that the Surveyor is occupied in inspecting or testing the material, or in travelling to and from the place of inspection to cover loss of time, subsistence, and other expenses.

Steel boilers not built under survey.

If no such inspection of materials is made during manufacture, the Surveyor is not to give a declaration in the case of steel boilers or machinery unless he has satisfied himself, by requiring a sufficient number of plates or quantity of material to be taken out and tested, or in some other effectual manner, that the material and workmanship are entirely

satisfactory, and in no case without a special reference to the Board of Trade.

143. In the case of steamers performing ocean voyages and coming in for survey, no question as to gear need be raised if the following spare gear and stores, or their equivalent, are supplied. The particulars of equivalent spare gear should be submitted to the Board for consideration. The heavier portions of this gear should be fitted and tried in their places, and should be kept on board where access can at all times be had to them:— Spare gear and stores to be carried.

1 pair of connecting rod brasses.
1 air-pump bucket, and rod, with guide.
1 circulating pump bucket and rod.
1 air-pump head-valve, seat, and guard.
1 set of india-rubber valves for air-pumps.
1 circulating pump head-valve, seat, and guard.
1 set of india-rubber valves for circulating pumps.
2 main bearing bolts and nuts.
2 connecting rod bolts and nuts.
2 piston rod bolts and nuts.
8 screw shaft coupling bolts and nuts.
1 set of piston springs suitable for the pistons.
3 sets, if of india-rubber, or 1 set if of metal, of feed-pump valves and seats.
3 sets, if of india-rubber, or 1 set if of metal, of bilge pump valves and seats.
1 hydrometer.
3 boiler tubes for each boiler.
100 iron assorted bolts, nuts, and washers, screwed but need not be turned.
12 brass bolts and nuts, assorted, turned and fitted.
50 iron bolts and nuts, assorted, turned, and fitted.

50 condenser tubes.

100 sets of packing for condenser tube ends, or an equivalent.

At least one spare spring of each size for escape valves.

1 set of water-gauge glasses.

$\frac{1}{10}$th of the total number of fire-bars necessary.

3 plates of iron, assorted.

6 bars of iron, assorted.

1 complete set of stocks, dies, and taps, suitable for the engines.

1 smith's anvil.

1 fitter's vice.

Ratchet-braces and suitable drills.

1 copper or metal hammer.

Suitable blocks and tackling for lifting weights.

1 dozen files, assorted, and handles for the same.

1 set of drifts or expanders for boiler tubes.

1 set of safety-valve springs (if so fitted) for every four valves; if there are not four valves, then at least one set of springs must be carried.

1 screw jack.

And a set of engineer's tools suitable for the service, including hammers and chisels for vice and forge; solder and soldering iron; sheets of tin and copper; spelter; muriatic acid, or other equivalent, &c., &c.

Size of shafting.

144. Main, tunnel, propeller, and paddle shafts should not be passed if less in diameter than that found by the following formulæ, without previously submitting the whole case to the Board of Trade for their consideration. It will be found that first-class makers generally put in larger shafts than those obtained by the formulæ.

For compound condensing engines with two or

more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}$$

$$P = \frac{f \times S^3}{C \times D^2}\left(2 + \frac{D^2}{d^2}\right)$$

Where S=diameter of shaft in inches.

d^2= square of diameter of high pressure cylinder in inches or sum of squares of diameters when there are two or more high pressure cylinders.

D^2=square of diameter of low pressure cylinder in inches or sum of squares of diameters when there are two or more low pressure cylinders.

P = *absolute* pressure in lbs. per square inch, that is, boiler pressure plus 15lbs.

C = length of crank in inches.

f = constant from following table.

For ordinary condensing engines, with one, two, or more cylinders, when the cranks are not overhung:—

$$S = \sqrt[3]{\frac{C \times P \times D^2.}{3 \times f}}$$

$$P = \frac{3 \times f \times S^3}{C \times D^2}$$

Where S=diameter of shaft in inches.

D^2= square of diameter of cylinder in inches or sum of squares of diameters when there are two or more cylinders.

P = *absolute* pressure in lbs. per square inch, that is, boiler pressure plus 15lbs.

C = length of crank in inches.

f = constant from following table:—

The presence of zinc in such boilers is also objectionable, and Surveyors should refuse to pass boilers so fitted for use on board vessels sailing under the provisions of Part III. of the Merchant Shipping Act, 1894.

When the water is pumped into the condenser there should be an efficient escape valve fitted to it, which cannot be readily tampered with, and if the condensing portion of the apparatus or the cooler and filter be unfit to bear the pressure on the boiler, an efficient safety-valve that cannot be readily over-loaded should be fitted between the steam pipe and the apparatus.

Pump for distilling apparatus to be fitted as fire engine.

146. It is advisable that the donkey engine for pumping water through the condenser be so fitted that it can be made available in case of emergency for extinguishing fire in any part of the ship; a leather hose, with suitable bends and conductors, should be supplied for this purpose.

Stores to be carried with distilling apparatus.

147. The following list of tools and materials should be provided for distilling apparatus:—

1 set of stoking tools.
1 scaling tool.
1 spanner for boiler doors.
1 set of fire bars, suitable for boiler.
1 14-inch flat bastard file.
1 14-inch half-round file.
1 10-inch round file.
3 file handles.
2 hand cold chisels.
1 chipping hammer.
1 pair of efficient gas tongs.
1 soldering iron.
10 lbs. of solder.
2 lbs. of rosin.
6 gauge glasses.

24 india-rubber gauge-glass washers.
30 bolts and nuts, assorted.
1 slide rod for donkey pump.
5 lbs. spun yarn.
10 lbs. cotton waste.
1 deal box, with lock, complete.
2 gallons machinery oil.
Animal charcoal sufficient to charge the filter at least twice.
1 can for machinery oil.
1 oil feeder.
1 small bench vice.
1 ratchet brace.
4 drills, assorted.
1 set dies and taps suitable for the bolts.
2 glass salinometers.
1 hydrometer and pot.
1 shifting spanner.
1 lamp for engineer.
And any other articles that the particular distiller and boiler supplier may, in the Surveyor's judgment, require.

148. List of distilling apparatus that have been approved by the Board of Trade.* List of approved distilling apparatus.

1. Normandy's Patent.
2. Winchester and Graveley's Patent.
3. Chaplin and Co.'s Patent.
4. Brown, A. and R. and Co.'s Patent.
5. Fraser's Patent.
6. Hocking and Co.'s Patent.
7. Hocking, Glascodine and Co.'s Patent.
8. Kirkaldy's Patent.
9. Siddeley and Co.'s Patent.
10. Scott, Son and Watts' Patent.

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*NOTE.—Any departure from the designs should be reported to the Board for consideration.

11. Smillie's Patent.
12. Rayner's Patent.
13. Russell's Patent.
14. Union S.S. Co.'s Patent.
15. Willoughby and Co.'s Patent.
16. Newton and Quiggin's Patent.
17. Row and Co.'s Patent.
18. Suffield and Brown's Patent.
19. Caird and Rayner's Patent.
20. Brown and Skinner's
21. Davie and Horne's.

Accidents and Damages.

Damage to be inspected.

149. In all cases in which a steam ship has sustained damage from any accident, or other cause, affecting her seaworthiness or efficiency, in any part of her hull, equipments, or machinery, the Surveyors of the port, where the vessel may be, are to go on board and ascertain the extent of the damage. In doing so they will take care not to make any change in the position or condition of things on board, which would be at all likely to affect any legal evidence, should an inquiry into the case afterwards be considered necessary.

It is extremely important that the vessel should be surveyed and a report sent to the Board as early as possible.

Report to be sent to Board of Trade.

150. The Surveyors are forthwith to send a report, together with a rough figured sketch, showing the position of all the bulkheads, and distinguishing the compartments that may have been injured, to the Board of Trade, in order that the certificate may be cancelled, if necessary. In this report they are to state whether the vessel is rendered inefficient or unseaworthy either in hull or machinery; and

whether, in their opinion, the certificate should be cancelled or only temporarily withdrawn.

Lights to be examined in cases of collision.

151. In cases of collision the Surveyor should examine the lights and screens of the vessel, and if the vessel with which she has been in collision is within his district he should examine the lights of that vessel also, and should forward to the Board of Trade, in a separate report, a statement respecting the size, condition, and place of her lamps and screens, and whether in his opinion they comply with the requirements of the regulations, and if not, in what respects they are deficient.

The Surveyors should be very careful in wording their reports as to lights and screens, as they may be used in evidence in cases taken into court.

Surveyor may order vessel to be docked.

152. If, in consequence of any accident to a steam ship, or for any other reason, the Surveyor considers it necessary to require the vessel to be taken into dock for the purpose of surveying the hull thereof, he may do so, but he is to be cautious never to exercise this power unless the circumstances of the case actually require it.

Certificate to be held during repairs.

153. Unless directions to the contrary are given by the Board of Trade, the Surveyor is to hold the certificate of the vessel during the time she is under repair, and for this purpose he is to require it of the master, unless the damage is such that in the Surveyor's judgment the certificate should be cancelled, in which case he should forward it to the Board of Trade with his report. *In all cases of serious damage the certificates should be returned to the Board of Trade to be cancelled.*

Certificate may be cancelled if necessary.

Should the master or owner decline to give up the certificate, the Surveyor is to send information immediately to the Board of Trade, so that, if necessary, it may be cancelled; but the Surveyor is

particularly directed to pay attention to the above instructions respecting reporting the whole case to the Board of Trade, as soon as possible after it comes to his notice.

Indorsement of certificate.

154. When the vessel is in every respect rendered seaworthy and efficient both in hull and machinery, to the entire satisfaction of the Surveyor, he is to put an indorsement on the back of the certificate and duplicate, if they have not been previously cancelled, to the effect that the damage has been made good so as to last for the period for which the certificate was granted, and return the certificate to the master or owner, or his agent, and allow the vessel to proceed. He is then to send to the Board of Trade a statement of the extent of the repair that was necessary, and a copy of the indorsement made on the certificate.

Penalty for not giving up cancelled or expired certificate.

155. In all cases in which a Surveyor is directed by the Board of Trade to obtain from an owner or master a certificate that has expired, or has been cancelled or revoked, he is to apply for it without delay, and in the event of his application not being attended to, he is at once to report the case. The 280th section of the Merchant Shipping Act, 1894, imposes a penalty of £10 on any owner or master who neglects or refuses to deliver up such certificate when required to do so by the Board of Trade.

List of Minutes embodied.

156. The substance of the following General and other Minutes has been embodied in the present edition of this Book of Instructions:—M. 13527 of 1901, 9716 of 1902, 1053, 7337, and 20918 of 1903, and 3577 and 8189 of 1904.

APPENDIX A.

CYLINDRICAL BOILER SHELLS.

JOINTS WITH DRILLED HOLES.

Formulæ for ordinary chain riveted and ordinary zig-zag riveted joints, and for joints of these descriptions, when every alternate rivet in the outer or in the outer and inner rows has been omitted:—

Let E = distance from edge of plate to centre of rivet in inches.

V = distance between rows of rivets in inches.

V_1 = distance between inner and middle row of rivets in inches for joints J. and K. (Figs. 18 and 15.)

B = boiler pressure in lbs. per square inch.

C = 1 for lap or single butt joints.

C = 1·75 for double butt joints.

d = diameter of rivets in inches.

D = inside diameter of boiler in inches.

F = factor of safety for shell plates, as by Clause 87 or Clause 103.

n = number of rivets in one pitch.

p_D = diagonal pitch in inches.

P_D = diagonal pitch in inches between inner and middle rows of rivets in inches for joint J.

p = greatest pitch of rivets in inches.

r = percentage of plate left between holes in greatest pitch.

R = percentage of value of rivet section.

R_1 = percentage of combined plate and rivet section.

S = tensile strength of material in lbs. per square inch of section.

S_1 = tensile strength of plates in tons.
T = thickness of plate in inches.
T_1 = thickness of each butt strap in inches.
% = least value of r, R, R_1 as the case may be, divided by 100.

When joints are used in boiler construction other than those shown in the attached sketches, or when any of the rivets are pitched less than two diameters apart, the particulars of such joints should be submitted for the consideration of the Board.

Ordinary Chain and Zig-zag Riveted Joints

Iron plates and iron rivets or steel plates and steel rivets:—

$$\frac{100\ (p-d)}{p.} = r$$

Iron plates and iron rivets:—

$$\frac{100 \times d^2 \times \cdot 7854 \times n \times C}{p \times T} = R.$$

Steel plates and steel rivets:—

$$\frac{100 \times 23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times p \times T} = R.$$

Given C, d, F, n, T, to find p, so that r and R are equal.

Iron plates and iron rivets:—

$$\frac{d^2 \times \cdot 7854 \times n \times C}{T} + d = p$$

Steel plates and steel rivets:—

$$\frac{23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times T} + d = p$$

Given C, F, n, T, r, to find p and d.

Iron plates and iron rivets:—

$$\frac{r \times T}{(100-r) \times \cdot 7854 \times n \times C} = d.$$

$$\frac{100 \times r \times T}{(100-r)^2 \times \cdot 7854 \times n \times C} = p$$

Steel plates and steel rivets:—

$$\frac{4 \cdot 5 \times S_1 \times r \times T}{23 \times (100-r) \times \cdot 7854 \times n \times C \times F} = d.$$

$$\frac{100 \times 4 \cdot 5 \times S_1 \times r \times T}{23 \times (100-r)^2 \times \cdot 7854 \times n \times C \times F} = p.$$

Iron plates and iron rivets, or steel plates and steel rivets when d is found first, then:—

$$\frac{100d}{100-r} = p.$$

Butt Straps.

Iron plates and iron butt straps or steel plates and steel butt straps:—

Double butt straps:—

$$\frac{5 \times T}{8} = T_1$$

Single butt straps:—

$$\frac{9 \times T}{8} = T_1$$

For Distance between Rows of Rivets, &c.

Iron and steel:—

$$\frac{3 \times d}{2} = E$$

Chain-riveted joints, Figs. 2, 4, 6, 9, 11, not less than:—

$$2 \times d = V.$$

See note page 312.

Zig-zag riveted joints, Figs. 3, 5, 7, 10, 12:—

$$\frac{\sqrt{(11p + 4d)\ (p + 4d)}}{10} = V.$$

Diagonal pitch, Figs. 3, 5, 7, 10, 12:—

$$\frac{6p + 4d}{10} = p_D$$

To Determine the Working Pressure.

$$\frac{S \times \% \times 2T}{F \times D} = B.$$

Chain and Zig-zag Riveted Joints in which every alternate Rivet has been omitted in the Outer Row, or in the Outer and the Inner Rows such as are shown by the Sketches on Page 318 and following.

Iron plates and iron rivets or steel plates and steel rivets:—

$$\frac{100\ (p - d)}{p} = r$$

Iron plates and iron rivets:—

$$\frac{100 \times d^2 \times \cdot 7854 \times n \times C}{p \times T} = R.$$

Steel plates and steel rivets:—

$$\frac{100 \times 23 \times d^2 \times \cdot 7854 \times n \times C \times F}{4 \cdot 5 \times S_1 \times p \times T} = R.$$

Iron plates and iron rivets or steel plates and steel rivets:—

$$\frac{100\ (p - 2d)}{p} + \frac{R}{n} = R_1$$

Butt Straps.

Where the number of rivets in the inner row is double the number in the outer row.

Iron-plates and iron butt-straps or steel plates and steel butt-straps.

Double butt-straps:—

$$\frac{5 \times T\ (p-d)}{8 \times (p - 2d)} = T_1$$

Single butt-straps:—

$$\frac{9 \times T\ (p-d)}{8 \times (p - 2d)} = T_1$$

When the number of rivets in the inner row is the same as in the outer row.

Double butt-straps:—

$$\frac{5 \times T}{8} = T_1$$

Single butt-straps:—

$$\frac{9 \times T}{8} = T_1$$

FOR DISTANCE BETWEEN ROWS OF RIVETS, &c.

Iron and steel:—

$$\frac{3 \times d}{2} = E.$$

Chain riveted joints, Figs. 13, 14, 15, 19:—

$$\frac{\sqrt{(11p + 4d)\ (p + 4d)}}{10} = V$$

or

$$2 \times d = V.$$

The *greater* of these two values of V to be used. *See note below.*

For joint K. (Fig 15):—

$2 \times d = V_1$ *See note on page* 312.

Zig-zag riveted joints, Figs. 16, 17, 18, 20:—

$$\sqrt{(\tfrac{11}{20} p + d)\ (\tfrac{1}{20} p + d)} = V.$$

Diagonal pitch, Figs. 16, 17, 18, 20:—

$$\tfrac{3}{10} p + d = P_D$$

For joint J. (Fig. 18):—

$$\frac{\sqrt{(11p + 8d)\ (p + 8d)}}{20} = V_1$$

Diagonal pitch (Fig. 18) :—

$$\frac{3p + 4d}{10} = P_D$$

To Determine the Working Pressure.

$$\frac{S \times \% \times 2T}{F \times D} = B.$$

NOTE.—The minimum value of V or V_1 for chain riveted joints is given as $2d$, but $\frac{4d + 1,}{2}$ is more desirable.

Maximum Pitches for Riveted Joints.

T = Thickness of plate in inches.

p = Maximum pitch of rivets in inches, provided it does not exceed 10 inches.

C = Constant applicable from the following table :—

Number of Rivets in one Pitch.	Constants for Lap Joints.	Constants for Double Butt Strap Joints.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

$$(C \times T) + 1\tfrac{5}{8} = p.$$

When the work is first class, such pitches may be adopted so far as safety is concerned, yet, in some cases, it may be well not to adopt the greatest pitch found by the formula. The maximum pitch should *not*, however, exceed 10 inches with the thickest plates for boiler shells. If in any case the pitch is found to exceed that arrived at by the foregoing formula, for the particular description of joint and thickness of plate, such pitches should *not* be passed, but all such cases should be reported.

Ordinary Chain & Zigzag Riveted Joints.

— Double Riveted Lap Joints. —

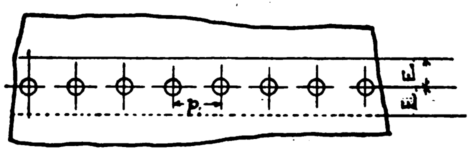


FIG. 1.

$$\frac{3d}{2} = E.$$

— Chain Riveting. —

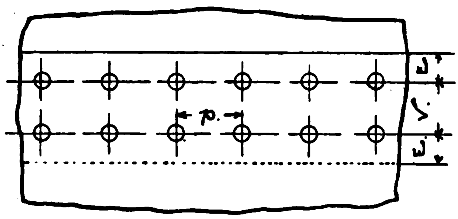


FIG. 2.

$$\frac{4d+1}{2} = V.$$

— Zigzag Riveting. —

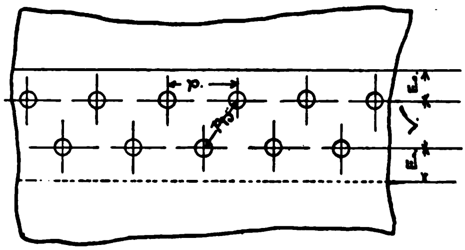


FIG. 3.

$$\frac{\sqrt{(11p+4d)(p+4d)}}{10} = V.$$

Ordinary Chain & Zigzag Riveted Joints.

— Treble Riveted Lap Joints. —

— Chain Riveting. —

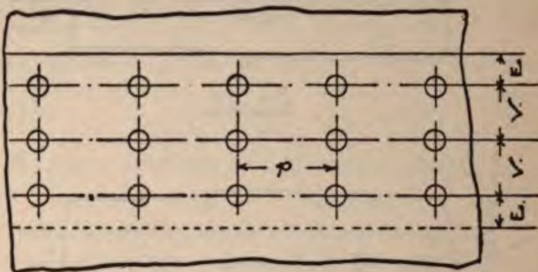


FIG. 4.

$$\frac{4d+1}{2} = V.$$

— Zigzag Riveting —

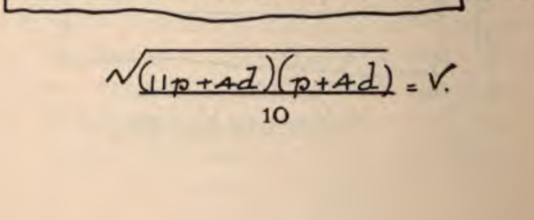


FIG. 5.

$$\frac{\sqrt{(11p+4d)(p+4d)}}{10} = V.$$

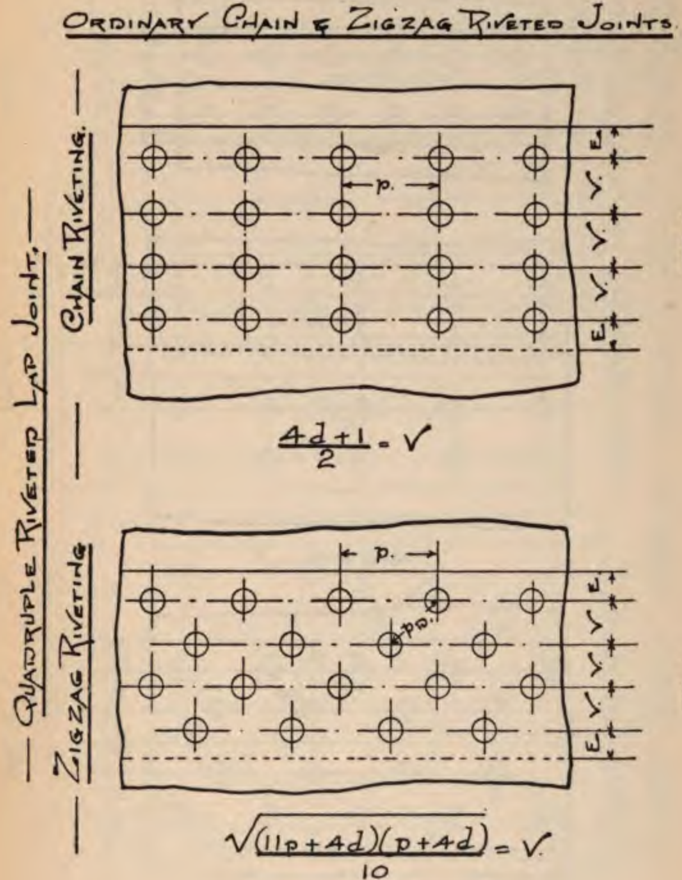

ORDINARY CHAIN & ZIGZAG RIVETED JOINTS.
— QUADRUPLE RIVETED LAP JOINT, —
— CHAIN RIVETING. —
— ZIGZAG RIVETING —
p.
E.
V.
V.
V.
E.
$\frac{4d+1}{2} = V$
p.
pd.
E.
V.
V.
V.
E.
$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V.$

FIG. 6.

FIG. 7.

Ordinary Chain & Zigzag Riveted Joints.

— Single Riveted —
— Double Butt Joint —

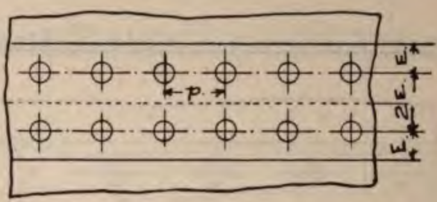


FIG. 8.

— Double Riveted Double Butt Joints. —
— Chain Riveting. —

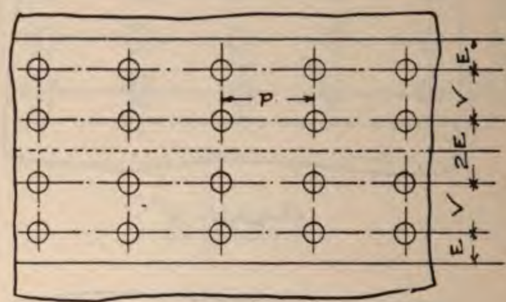


FIG. 9.

$$\frac{4d+1}{2} = V.$$

— Zigzag Riveting —

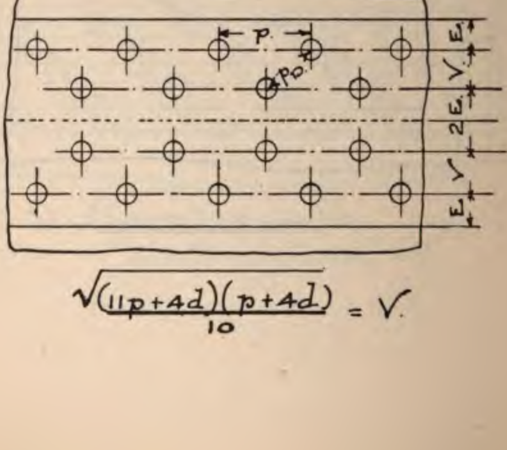


FIG. 10.

$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V.$$

Ordinary Chain & Zigzag Riveted Joints.

— Treble Riveted Double Butt Joints. —

— Chain Riveting. —

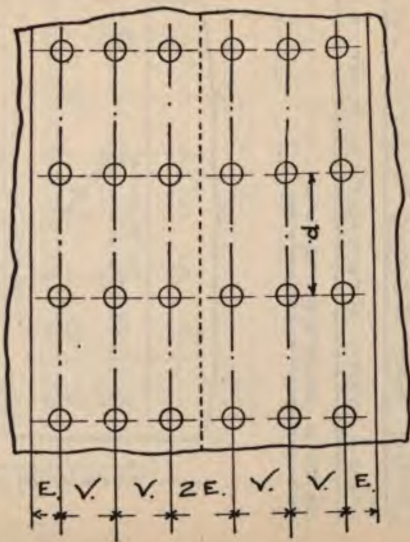


FIG. 11.

$$\frac{4d+1}{2} = V$$

— Zigzag Riveting. —

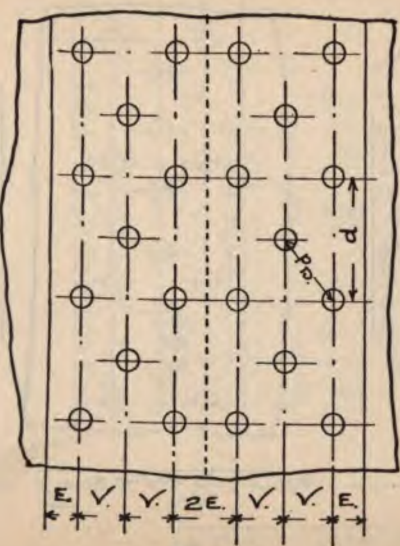


FIG. 12.

$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V$$

CHAIN AND ZIGZAG RIVETED JOINTS IN WHICH EVERY ALTERNATIVE RIVET IS OMITTED IN THE OUTER ROWS.

— TREBLE RIVETED LAP JOINT. —

— CHAIN RIVETING. —

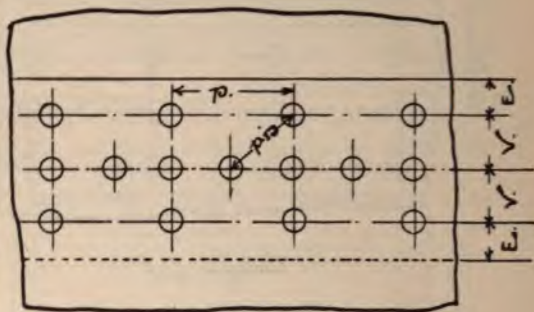


FIG. 13.

$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V$$

— ZIGZAG RIVETING —

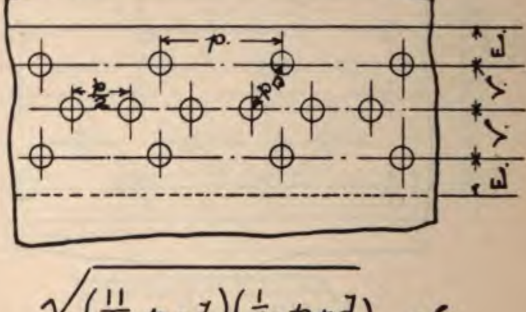


FIG. 16.

$$\sqrt{\left(\tfrac{11}{20}p+d\right)\left(\tfrac{1}{20}p+d\right)} = V$$

DOUBLE RIVETED DOUBLE BUTT JOINTS.

EACH ALTERNATE RIVET OMITTED IN THE OUTER ROW

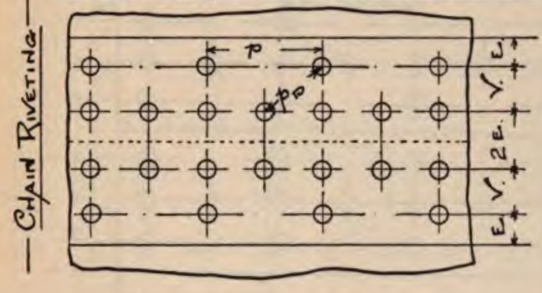
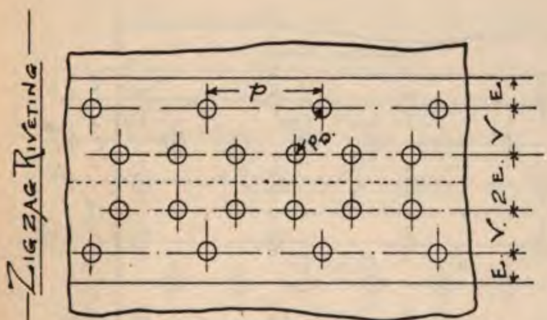


FIG. 19.

$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V$$



$$\sqrt{(\tfrac{11}{20}p+d)(\tfrac{1}{20}p+d)} = V$$

FIG. 20.

TREBLE RIVETED DOUBLE BUTT JOINT.

EACH ALTERNATE RIVET OMITTED IN OUTER & INNER ROWS

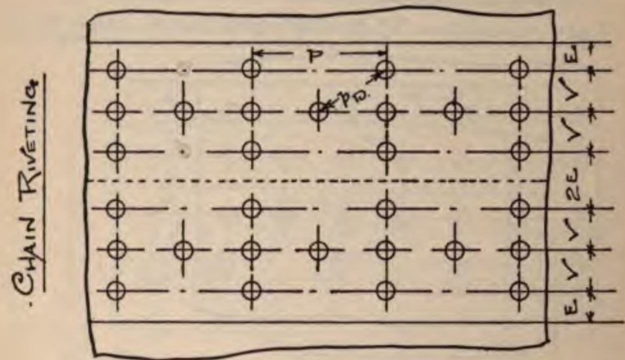


FIG. 14.

$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V$$

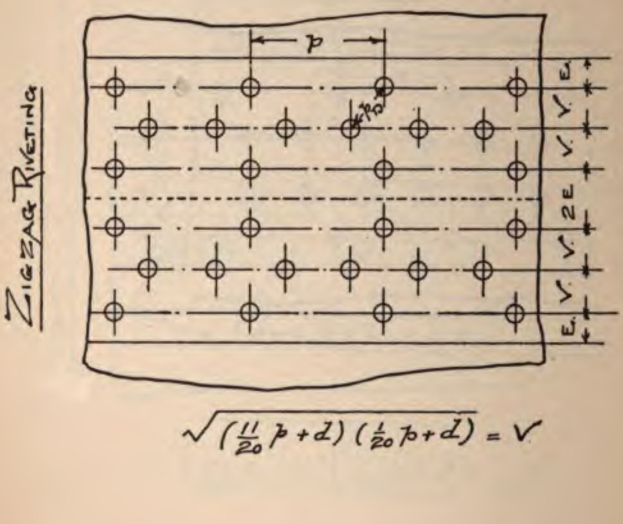


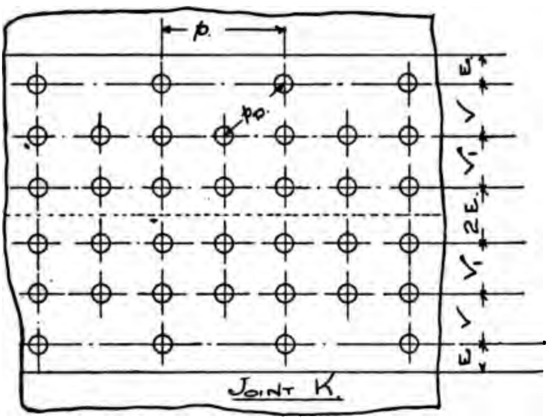
FIG. 17.

$$\sqrt{\left(\frac{11}{20}p+d\right)\left(\frac{1}{20}p+d\right)} = V$$

— TREBLE RIVETED DOUBLE BUTT JOINT —

EACH ALTERNATE RIVET OMITTED IN THE OUTER ROW.

— CHAIN RIVETING. —



$$\sqrt{\frac{(11p+4d)(p+4d)}{10}} = V$$

$$\frac{4d+1}{2} = V_1$$

$$\frac{3 \times d}{2} = E'$$

FIG. 15.

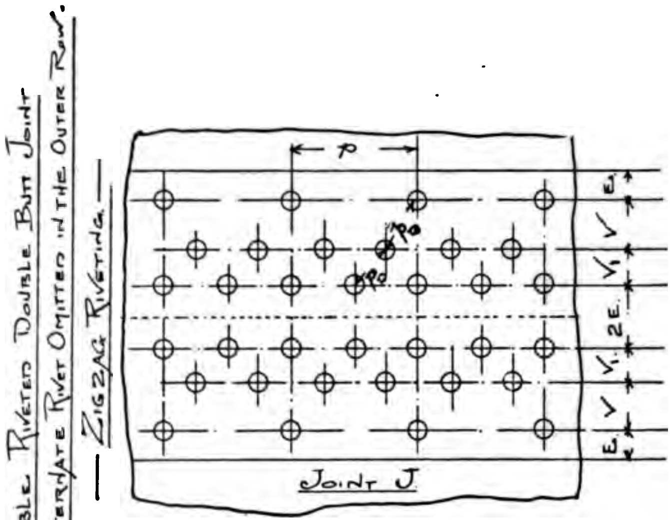


FIG. 18.

$$\sqrt{\left(\tfrac{11}{20}p+d\right)\left(\tfrac{1}{20}+d\right)} = V$$

$$\sqrt{\frac{(11p+8d)(p+8d)}{20}} = V_1$$

$$\frac{3p+4d}{10} = P_D$$

$$\tfrac{3}{10}p + d = p_D$$

APPENDIX B.

EXPLANATORY AND SUPPLEMENTARY INSTRUCTIONS RESPECTING THE TESTING OF STEAM PIPES.

See also Clause 116.

MAIN STEAM PIPES.

1. (*a*). All copper pipes having brazed longitudinal seams, whether forming a complete range or only part of a range of pipes, should, with the exception of those referred to in paragraph (*c*), be examined and tested with the lagging removed at least once in about every four years.

(*b*). Iron, steel, or solid drawn copper pipes should, when the diameter exceeds 2½ inches, be stripped and tested by hydraulic pressure at least once in six years.

(*c*). In the case of pipes of 2½ inches diameter, or less, the Surveyor may use his own discretion as to the removal of the lagging for more than a few inches near each flange when the hydraulic test is applied.

AUXILIARY STEAM PIPES HAVING AN INTERNAL DIAMETER EXCEEDING 6 INCHES.

2. (*a*). Copper pipes having brazed longitudinal seams should be stripped and tested by hydraulic pressure at least once in four years.

(*b*). Iron, steel, or solid drawn copper pipes should be stripped and tested by hydraulic pressure at least once in six years.

AUXILIARY STEAM PIPES HAVING AN INTERNAL DIAMETER OF OVER 3 INCHES AND NOT EXCEEDING 6 INCHES.

3. (*a*). Copper pipes having brazed longitudinal seams should be stripped for not less than 2 inches at each flange and tested by hydraulic pressure at least once in every four years.

(*b*). Iron, steel, or solid drawn copper pipes should be stripped for not less than 2 inches at each flange and tested by hydraulic pressure at least once in six years.

Small Auxiliary Steam Pipes and Auxiliary Steam Pipes Situated in Open Spaces.

4. Steam pipes not exceeding 3 inches diameter and steam pipes situated on or above the upper deck or other open spaces should be tested periodically, but the Surveyor may exercise his discretion as to when the test shall be made, and as to whether the pipes shall be wholly stripped or stripped beyond a small portion adjacent to the flanges when the hydraulic test is applied.

General

5. In all cases where the pipes are not wholly stripped, the hydraulic test pressure should remain on the pipes for such time as the Surveyor considers necessary, but in no case for less than 20 consecutive minutes. Any length from which leakage is observed at other places than the flanges should be stripped, repaired, and re-tested.

6. The test pressure to be applied in all cases should be not less than twice the maximum working pressure to which the pipes are subjected.

7. When a vessel is surveyed for a passenger certificate after transference from a foreign flag, or for the first time, all the steam pipes should be tested as indicated above.

8. All main steam pipes, new, old, or repaired, whatever their size may be, and all auxiliary steam pipes over 3 inches in internal diameter should be tested in the presence of the Surveyor, but in the case of auxillary steam pipes not exceeding 3 inches in diameter, the Surveyor, if he cannot conveniently be present at the tests on account of other duties, should make such arrangements as will ensure the

pipes being tested by the makers or repairers in accordance with the regulations, and satisfy himself that the tests have been properly carried out.

9. Surveyors should see that the thickness of pipes, provision for expansion, adjustment of the guard bolts, drainage, &c., are in all respects in compliance with the Board's regulations.

10. Main steam pipes include the main range and its branches from the various boilers and those to the propelling machinery, also all steam pipes joining two or more boilers together.

11. When a range of pipes is tested by hydraulic pressure, the Surveyor should exercise his discretion as to the taking down of a few of the pipes in order that their interiors may be examined and the actual thickness ascertained.

12. In addition to the periodical examinations enumerated above, the Surveyor may at any time he thinks it necessary, in order to satisfy himself as to any doubtful part or parts, require the main or auxiliary pipes to be tested during the survey of a vessel, and may also require the removal of any of such pipes to ascertain their thickness or condition.

13. The above instructions apply to all steam pipes, the bursting of which would propably cause loss of life or serious injury, but it is not expected that the Surveyors will insist on the testing of small pipes, the free outflow of steam from which would cause no danger or inconvenience, and which would be difficult to burst under any circumstances.

14. In the case of vessels, new or old, in which it is proposed to fit arrangements for superheating the steam, full particulars of the proposed superheater and steam pipes should be submitted for consideration with as little delay as possible, and in no case should a declaration be issued until the arrangement has been sanctioned.

LLOYD'S RULES FOR THE SURVEY AND CONSTRUCTION OF ENGINES AND BOILERS OF STEAM VESSELS.

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1. In steam vessels, the machinery and boilers are to be inspected throughout construction, the boilers tested by hydraulic pressure, and the machinery tested under steam by the Society's Engineer-Surveyors, who will furnish a report to the Committee describing them in the manner shown in form No. 8. If found satisfactory, the Committee will thereupon grant a certificate, and insert in the Register Book the notification, "LMC" *in red* (*i.e.* "LLOYD'S MACHINERY CERTIFICATE"), indicating that the machinery and boilers are certified to be in good order and safe working condition.

Special Survey of New Engines or Boilers.

2. In steam vessels built under Special Survey, the Machinery and Boilers must also be constructed under Special Survey.

3. In cases of machinery or new boilers being built under Special Survey, the distinguishing mark ✠ will be noted in red, thus: "✠ LMC," or "✠ NE & B," or "✠ NB."

4. In order to facilitate this inspection, the plans of the machinery and boilers are to be examined and from them the working pressure fixed.

5. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under steam; any defects, etc., to be pointed out as early as possible.

6. The Surveyors may also, if desired, compare the work as it progresses with the requirements of the specification agreed upon by the parties concerned, and certify to the conditions thereof, as far as can be seen, being satisfactorily complied with.

Boilers.

7. The Surveyors will be guided in fixing the working pressure by the tables and formulæ annexed. (*See* paragraph 41.)

8. Any novelty in the construction of the machinery or boilers to be reported to the Committee.

9. The boilers, together with the machinery, to be inspected at different stages of construction.

All the holes in steel boilers should be drilled, but if they be punched the plates are to be afterwards annealed.

All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed.

No steel stays are to be welded.

Unless otherwise specified, the Rules for the construction of iron boilers will apply equally to boilers made of steel.

10. The boilers to be tested by hydraulic pressure, in the presence of the Engineer-Surveyor, to twice the working pressure, and carefully gauged while under test.

11. Two safety valves to be fitted to each boiler, and loaded to the working pressure in the presence of the Surveyor. In the case of boilers of greater working pressure than 60lbs. per square inch, the safety valves may be loaded to 5lbs. above the working pressure. If common valves are used their combined areas to be at least half a square inch to each square foot of grate

surface. If improved valves are used, they are to be tested under steam in the presence of the Surveyor; the accumulation in no case to exceed 10 per cent. of the working pressure.

12. An approved safety valve also to be fitted to the super-heater.

13. In winch boilers one safety valve will be allowed, provided its area be not less than half a square inch per square foot of grate surface.

14. Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

15. Stop-valves to be fitted so that each boiler can be worked separately.

16. Each boiler to be fitted with a separate steam gauge, to accurately indicate the pressure.

17. Each boiler to be fitted with a blow-off cock independent of that on the vessel's outside plating.

18. The machinery and boilers are to be securely fixed to the vessel to the satisfaction of the Surveyor.

Quality and Testing of Boiler Steel.

19. When steel is used in the construction of boilers intended for vessels classed or proposed for classification in the Society's Register Book, the boilers shall be constructed in accordance with the requirements of the Rules, and the following conditions be fulfilled:—

1. **Process of Manufacture.**—Steel for Marine Boilers shall be made by the Open Hearth process acid or basic.

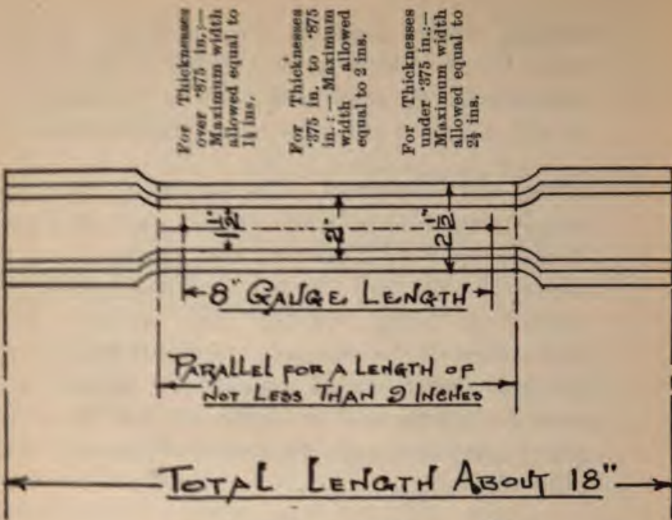
2. **Freedom from Defects.**—The finished material shall be free from cracks, surface flaws, and lamination. It shall also have a workmanlike finish, and must not have been hammer-dressed.

3. **Testing and Inspection.**—The following tests and inspections shall be made at the place of manufacture prior to despatch; but, in the event of any of the material proving unsatisfactory in the course of being worked into boilers, such material shall be rejected, notwithstanding any previous certificate of satisfactory testing, and such further tests of the material from the same charge may be made as the Surveyor may consider desirable.

4. **Tensile Test Pieces.**—The tensile strength and ductility shall be determined from Standard test pieces cut lengthwise or crosswise from the rolled material. When material is annealed or otherwise treated before despatch, the test pieces shall be similarly and simultaneously treated with the material before testing.

Plates, Angles and Tee Bars.—Wherever practicable the rolled surfaces shall be retained on two opposite sides of the test piece. The elongation shall be measured on a Standard test piece having a gauge length of 8 inches.

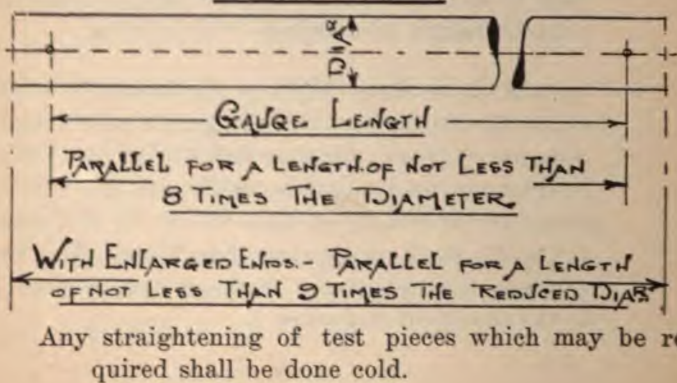
For material more than ·875in. in thickness the width of the test piece between the gauge points shall not exceed 1½ins.; for material ·875in. to ·375in. in thickness, inclusive, the width shall not exceed 2ins.; for material less than ·375in. in thickness the width shall not be more than 2½ins. In other respects the test pieces shall conform generally to the Standard test piece A.



— TEST PIECE A —

Round bars may be tested either full size as rolled, or turned down when the diameter is considerable. The test piece shall have a gauge length of not less than 8 times its diameter, and a sectional area of not less than ¼ square inch. When enlarged ends are used the length of the parallel portion shall not be less than 9 times the reduced diameter, *see* Standard test piece B.

— TEST PIECE B —



Any straightening of test pieces which may be required shall be done cold.

5. **Mechanical Tests, and Selection of Test Pieces.**—Plates and bars for boilers shall comply with the following mechanical tests. All test pieces shall be selected by the Surveyor and tested in his presence, and he shall satisfy himself that the conditions herein described are fulfilled.

6. **Tensile Tests.**—Plates:—The tensile breaking strength of steel plates for shells and girders, determined from Standard test pieces, shall be between the limits of 28 and 32 tons per square inch. For plates intended for flanging or welding, and for combustion chambers and furnaces, the tensile breaking strength shall be between the limits of 26 and 30 tons per square inch. In the case of material for purposes in which tensile strength is not important, the tensile test may be dispensed with and the bend test only be made, if so specified by the builders and approved by the Committee. The elongation, measured on a Standard test piece having a gauge length of 8ins., shall not be less than 20 per cent. for material of ·375in. in thickness and upwards required to have a tensile breaking strength of 28 to 32 tons per square inch; and not less than 23 per cent. for material of ·375in. in thickness and upwards required to have a tensile breaking strength of 26 to 30 tons per square inch.

Stay, Angle, and Tee Bars.—The tensile breaking strength of longitudinal stays and angle and tee bars shall be between the limits of 28 and 32 tons per square inch, with an elongation of not less than 20 per cent. of the gauge length measured on the Standard test pieces A or B. For bars for combustion chamber stays the tensile breaking strength shall be between 26 and 30 tons per square inch, with an elongation of not less than 23 per cent, of the gauge length measured on the Standard test piece B.

For material under ·375in. in thickness the elongation may be not more than 3 per cent. below the above-named elongations.

Rivet Bars:—The tensile breaking strength of rivet bars shall be between the limits of 26 and 30 tons per square inch of section, with an elongation of not less than 25 per cent. of the gauge length measured on the Standard test piece B. The bars may be tested the full size as rolled.

7. **Number of Tensile Tests.**—Plates.—One tensile test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one tensile test shall be taken from each end.

Angle, Tee, Rivet and Stay Bars.—At least two tensile tests for angle bars, tee bars, rivet bars, and stay bars shall be taken from each charge; but when the number of the bars, as rolled, from one charge exceeds 15, an additional tensile test shall be made for each further batch of 15 bars or portion thereof.

Should a tensile test piece break outside the middle half of its gauge length, and the elongation be less than that required by the Rules the test may, at the Maker's option, be discarded and another test be made of the same plate or bar.

8. **Bend Tests.**—Cold Bends.—Test pieces shall be sheared lengthwise or crosswise from plates or bars, and shall not be less than 1½ins. wide, but for small bars the whole section may be used. For rivet bars bend tests are not required.

Temper Bends:—The test pieces shall be similar to those used for cold bend tests. For temper bend tests the samples shall be heated to a blood red and quenched in water at a temperature not exceeding 80° Fahr. The colour shall be judged indoors in the shade.

In all cold bend tests, and in temper bend tests on samples ·5in. in thickness and above, the rough edge or arris caused by shearing may be removed by filing or grinding, and samples 1in. in thickness and above may have the edges machined, but the test pieces shall receive no other preparation. The test pieces shall not be annealed unless the material from which they are cut is similarly annealed, in which case the test pieces shall be similarly and simultaneously treated with the material before testing.

For both cold and temper bends the test piece shall withstand, without fracture, being doubled over until the internal radius is equal to 1½ times the thickness of the test piece, and the sides are parallel.

For small sectional material these bend tests may be made from the flattened bar.

Bend tests may be made either by pressure or by blows.

9. **Number of Bend Tests.**—Plates:—One cold or temper bend test shall be taken from each plate as rolled. For plates exceeding 2½ tons in weight one bend test shall be taken from each end—one bend test to be temper and the other cold.

Angle and Stay Bars:—A cold or temper bend test shall be made from each angle or stay bar rolled.

10. **Tests for Manufactured Rivets.**—Rivets selected by the Surveyor from the bulk shall withstand the following tests:—

(*a*) The rivet shanks are to be bent cold, and hammered until the two parts of the shank touch in the manner shown in Fig. 1, without fracture on the outside of the bend.

(*b*) The rivet heads are to be flattened, while hot, in the manner shown in Fig. 2, without cracking at the edges. The heads are to be flattened until their diameter is $2\frac{1}{2}$ times the diameter of the shank.

—Fig. 1.—

— Fig. 2.—

11. **Additional Tests before Rejection.**—Should any of the test pieces first selected by the Surveyor not fulfil the test requirements, two further tests may be made from the same plate or bar, but should either of these fail, the plate or bar from which the test pieces were cut shall be rejected. In all such cases further tests shall be made before any material from the same charge can be accepted.

12. **Branding.**—Every plate and bar shall be clearly and distinctly marked by the Maker in two places with the Society's brand, thus:— indicating that the material has complied with the Society's tests.

No plates or bars bearing this brand shall be forwarded from the Steel Works until the prescribed tests have been made by the Surveyor, and the mill sheets have been signed by him. All plates and bars shall also be legibly stamped in two places with the Maker's name or trade mark, and the place where made. They shall also be stamped with numbers or identification marks by

which they can be traced to the charge from which the material was made.

13. **Maker's Certificate.**—Before the mill sheets are signed by the Surveyor, the Maker shall furnish him with a certificate guaranteeing that the material has been made by the Open Hearth process, and that it has been subjected to, and has withstood satisfactorily, the tests above described in the presence of the Surveyor. The following form of certificate will be accepted if printed on each mill sheet with the name of the firm, and initialled by the Test House Manager:—

"We hereby certify that the material described below has been made by the Open Hearth process, and is that which has been satisfactorily tested in the presence of the Surveyor in accordance with the Rules of Lloyd's Register."

14. **Defacing of Rejected Material.**—In the event of the material failing, in any case, to withstand the prescribed tests, the Surveyor shall see that the Society's brand stamped on the plates and bars by the Maker has been defaced by punch marks extending beyond the brand in the form of a cross, thus:—

denoting that the material has been rejected.

15. **Facilities for Inspection.**—The Maker shall adopt a system of marking the ingots, billets, slabs, plates, bars, etc., which will enable all finished material to be traced to the original charge, and the Surveyor must be given every facility for tracing all plates and bars to their respective charges, and for witnessing the required tests. When he is satisfied with the material and with the results of the tests, he shall be furnished with

two copies of the advice notes of the material for his signature, one of which is to be forwarded by the manufacturer to the Boiler Maker, and the other is to be forwarded by the Surveyor to the Surveyors at the port where the boiler is to be built.

16. **Steel not produced where Rolled.**—Where steel is not produced in the works at which it is rolled, a certificate shall be supplied to the Surveyor, stating the Open Hearth process by which it was made, the name of the Steel Maker who supplied it, also the numbers of the charges for reference to the books of the Steel Maker. The number of the charge shall be marked on each ingot or billet for the purpose of identification, and the finished plates and bars shall also be legibly stamped in two places with the Maker's name or trade mark, and the place where made. They shall also be stamped with numbers or identification marks by which they can be traced to the charge from which the material was made.

17. **General.**—Besides the foregoing tests, samples of all material may be subjected to additional tests at the discretion of the Surveyors.

Engines.

20. The engines are to be fitted with two feed-pumps, each capable of supplying the boilers; the pumps, etc., to be so arranged that either can be overhauled whilst the other is at work.

21. The engines are to be fitted with two bilge-pumps, which are to be so arranged that either can be overhauled whilst the other is at work.

22. In engines of 70 H.P. and under, one feed-pump and one bilge-pump will be deemed sufficient, provided they are of adequate capacity.

The main feed-pumps may be worked by independent engines provided they are fitted with automatic regulators for controlling their speed. If only one such pump is fitted for the main feed, the auxiliary feed pump required by paragraph 25 should also be fitted with an automatic speed regulator.

23. A bilge injection, or a bilge suction to the circulating pump, is to be fitted.

24. The engine bilge-pumps are to be fitted capable of pumping from each compartment of the vessel, the peaks excepted. All bilge suction pipes are to be fitted with strum boxes or strainers, so constructed that they can be cleared without breaking the joints of the suction pipes. The total area of the perforations in the strainers should be not less than double that of the cross section of the suction pipe. The mud boxes and roses in engine room are to be placed where they are easily accessible, and to the satisfaction of the Surveyor.

25. A steam pump is to be provided capable of supplying the boilers with water; this pump to be provided with suctions to the hotwell and also to the sea. A steam pump is to be so fitted as to pump from each compartment, to deliver water on deck, and if no hand pump is fitted in engine room it must be fitted to be worked by hand. In small vessels in which only one steam pump is fitted, it must comply with all the requirements.

26. In all steam pipes provision is to be made for expansion and contraction to take place without unduly straining the pipes, and all main steam pipes are to be tested by hydraulic pressure to twice the working pressure, in the presence of the Engineer Surveyor.

27. All discharge-pipes to be, if possible, carried above the deep load-line, and to have discharge valves fitted on the plating of the vessel in an accessible position.

28. No pipes are to be carried through the bunkers without being properly protected.

29. Bilge suction-pipes to be arranged to pump direct from each compartment, the roses to be fixed in places where they can be easily accessible.

Shafts.

30. All shafts are to be made of good material, are to be examined when rough turned and when finished. In the case of screw shafts scrap steel is not to be used. It is recommended that these be made of ingot steel or forged from blooms made from rolled iron bar of good fibrous quality.

A tensile and a bend test are to be made on pieces cut from one end of each ingot steel shaft forging, the piece from which they are cut being of the same size as the body of the forging. In the case of built crank shafts the tests are to be taken from the material of the crank pins and journals, not from the webs. If more than one piece is forged from one ingot, one test only will be required from the ingot. The tensile strength is not to be less than 27 tons per square inch nor to exceed 32 tons per square inch. The elongation is not to be less than 30 per cent. in a length of two inches measured on a plain portion turned not more than three-quarters of an inch diameter. The bend test piece is to be made one inch square and must be capable of being bent cold without fracture, through an angle of 180° over a radius not greater than half-an-inch.

31. Gauges of an approved description for testing the truth of the crank shafts are to be supplied with all new engines, and adjusted in the presence of the Surveyor.

32. The length of the stern bush is to be at least four diameters of the shaft. It is recommended that the shaft liner should be continuous the whole length of the stern tube, and that the after end should be tapered in thickness and made watertight in the propeller boss. If the liner is made in two pieces the joint should be burned. If the liner does not fit tightly at the part between the

bearings in the stern tube, the space between the shaft and the liner should be charged or "forced" with a plastic material insoluble in water and non-corrosive. If two liners are used, it is recommended that they be tapered in thickness at the ends, and that the shaft should be lapped or protected between the liners. In this case, and also if no liners are used, the diameter of the shaft should be $\frac{21}{20}$ths of that required for a shaft with a continuous liner.

For dimensions of shafts, see the formula in paragraph 60.

Steel Castings.

33. **Steel Castings** may be used for engine purposes provided they fulfil the Committee's requirements, which are as follow:—

1. For purposes for which **Cast Iron** is ordinarily employed, such as propeller bosses and blades, bed plates, engine framing and columns, brackets, weigh-shaft levers, pistons, cylinder covers, eccentric straps, bearing bushes, etc., the castings must be sound, and are to be subjected to such drop and hammering tests as are practicable.
2. For shafts or parts of shafts, and for purposes for which **forgings** are ordinarily employed, the material must also be subjected to the following tests:—
3. A tensile test is to be made from a piece taken from each casting. The tensile strength is not to exceed 30 tons per square inch, and the elongation is not to be less than 10 per cent. in a length of 8 inches, and a cold bending test, turned to $1\frac{1}{4}$ inches diameter or planed to $1\frac{1}{4}$ inches square, is to be capable of being bent without fracture through an angle of 90° over a radius not greater than $1\frac{3}{4}$ inches.
4. All steel castings are to be thoroughly annealed.

Cocks, Pipes, and Sea Connections.

34. With a view to insuring a better control over cocks, valves, and pipes connecting the engines and boilers with the sea, they are to be fixed as follows, in all new vessels and vessels having *new engines or boilers*:—

35. All sea-cocks to be fitted on the plating of the vessel above the level of the stoke-hold and engine-room platforms, or attached to Kingston valves of a height sufficient to lift them up to the level of these platforms.

36. The bolts securing all cocks or sea connections to the plating of the vessel are to be tapped into the plating of the vessel or fitted with countersunk heads.

37. The blow-off cocks on the plating of the vessel are to be fitted with spigots passing through the plating, and a brass or gun-metal ring on the outside. The cocks are to be so constructed that the key or spanner can only be taken off when the cock is shut.

38. Cocks and valves connecting all suction pipes to be fixed above the stoke-hold and engine-room platforms.

39. The arrangements of pumps, bilge injections, suction and delivery pipes, is to be such as will not permit of water being run from the sea into the vessel by an act of carelessness or neglect. Any defective arrangement to be reported to the Committee.

Spare Gear.

40. The articles of spare gear mentioned in the following list will be required to be carried in all steam vessels classed in the Society's Register Book, viz.:—

2 connecting rod or piston rod top-end bolts and nuts
2 connecting rod bottom-end bolts and nuts.
2 main bearing bolts.
1 set of coupling bolts.
1 set of feed and bilge pump valves.
1 set of piston springs (where common springs are used).
A quantity of assorted bolts and nuts.
Iron of various sizes

In addition to the foregoing the following articles are recommended to be carried with a view to expedite repairs and lessen delay in distant ports, viz.:—

Crank shaft.
Propeller shaft.
Propeller, or a full set of blades.
Stern bush, or lignum vitæ lining for bush.
1 pair of connecting rod brasses.
1 pair of cross head brasses.
1 set of link brasses.
1 eccentric strap complete.
Air pump rod.
Circulating pump rod.
H.P. valve spindle.
L.P. valve spindle.
1 set of check valves.
6 cylinder cover bolts.
6 junk ring bolts.
4 valve chest cover bolts.
2 dozen boiler tubes.
3 dozen condenser tubes.
1 cylinder escape valve and spring.
1 set of safety valve springs.

Rules for Determining the Working Pressure to be Allowed in New Boilers.

Cylindrical Shells of Iron Boilers.

41. The strength of circular shells of iron boilers to be calculated from the strength of the longitudinal joints by the following formula:—

$$\frac{C \times T \times B}{D} = \text{working pressure.}$$

where C = co-efficient as per following table,
T = thickness of plate in inches,
D = mean diameter of shell in inches,
B = percentage of strength of joint found as follows—the least percentage to be taken.

For plate at joint $B = \dfrac{p-d}{p} \times 100.$

For rivets at joint $B = \dfrac{n \times a}{p \times T} \times 100$ with iron rivets in iron plates with punched holes.

For rivets at joint $B = \dfrac{n \times a}{p \times T} \times 90$ with iron rivets in iron plates with drilled holes.

(In case of rivets being in double shear, $1 \cdot 75a$ is to be used instead of a.)

where p = pitch of rivets,
d = diameter of rivets,
a = sectional area of rivets,
n = number of rows of rivets.

MEM.—In any case where the strength of the longitudinal joint is satisfactorily shown by experiment to be greater than given by this formula the actual strength may be taken in the calculation.

Table of Co-Efficients.

IRON BOILERS.

Description of Longitudinal Joint.	For Plates ½-inch thick and under.	For Plates ¾-inch thick and above ½-inch.	For Plates above ¾-inch thick.	
Lap Joint, Punched Holes	155	165	170	
Lap Joint, Drilled Holes	170	180	190	
Double Butt Strap Joint, Punched Holes ..	170	180	190	
Double Butt Strap Joint, Drilled Holes	180	190	200	

NOTE.—The inside butt strap to be at least ¾ of the strength of the longitudinal joint.

Cylindrical Shells of Steel Boilers.

42. The strength of cylindrical shells of steel boilers is to be calculated from the following formula:—

$$\frac{C \times (T - 2) \times B}{D} = \text{working pressure in lbs. per sq. in.}$$

where D = mean diameter of shell in inches.

T = thickness of plate in sixteenths of an inch.

C = 21 when the longitudinal seams are fitted with double butt straps of equal width.

C = 20·25 when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets.

C = 19·5 when the longitudinal seams are lap joints.

If the minimum tensile strength of shell plates is 28 or 29 tons per square inch instead of 27 tons per square inch these values of C may be correspondingly increased.

B = the least percentage of strength of longitudinal joint found as follows:—

For plate at joint $B = \frac{p - d}{d} \times 100$

For rivets at joint $B = \frac{n \times a}{p \times t} \times 85$ where steel rivets are used

$B = \frac{n \times a}{p \times t} \times 70$ where iron rivets are used.

where p = pitch of rivets in inches.

t = thickness of plate in inches.

d = diameter of rivet holes in inches.

n = number of rivets used per pitch in the longitudinal joint.

a = sectional area of rivet in square inches.

In case of rivets in double shear $1 \cdot 75a$ is to be used instead of a.

NOTE.—The inside butt strap to be at least $\frac{3}{4}$ of the strength of the longitudinal joint.

NOTE.—For the shell plates of superheaters or steam chests enclosed in the uptakes or exposed to the direct action of the flame, the co-efficients should be $\frac{2}{3}$ of those given in the preceding tables.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

Stays.

43. The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely:—

44. **Iron Stays.**—For stays not exceeding $1\frac{1}{2}$ inches smallest diameter, and for all stays which are welded 6,000 lbs. per square inch; for unwelded stays above $1\frac{1}{2}$ inches smallest diameter, 7500lbs. per square inch.

45. **Steel Stays.**—For **screw** stays not exceeding $1\frac{1}{2}$ inches smallest diameter, 8000lbs. per square inch; for **screw** stays above $1\frac{1}{2}$ inches smallest diameter, 9000lbs. per square inch. For other stays not exceeding $1\frac{1}{2}$ inches smallest diameter, 9000lbs. per square inch, and for stays exceeding $1\frac{1}{2}$ inches smallest diameter, 10000lbs. per square inch. No steel stays are to be welded.

46. **Stay Tubes.**—The stress is not to exceed 7500lbs. per square inch.

Flat Plates.

47. The stress of flat plates supported by stays is to be taken from the following formula:—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per sq. in.};$$

where T = thickness of plate in 16ths of an inch,

P^2 = square of pitch in inches. If the pitch in the rows is not equal to that between the rows, then the mean of the squares of the two pitches is to be taken.

C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads.

C = 100 for iron or steel plates above $\frac{7}{16}$ thick fitted with screw stays with riveted heads,

C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts.

C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts.

C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts,

C = 140 for iron plates fitted with stays with double nuts,

C = 150 for iron plates fitted with stays with double nuts and washers outside the plates, of at least $\frac{1}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,

C = 160 for iron plates with stays with double nuts and washers riveted to the outside of the plates, of at least $\frac{2}{5}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates.

C = 175 for iron plates fitted with stays with double nuts and washers riveted to the outside of the plates, when the washers are at least $\frac{3}{5}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{2}{3}$ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those for combustion chambers, the values of C may be increased as follows:—

C = 140	increased to	175,
150	„	185,
160	„	200,
175	„	220,
190	„	240.

48. If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times (T + \frac{t}{2})^2}{P^2} = \text{working pressure in lbs. per sq. in.};$$

where t = thickness of doubling plates in sixteenths, and C, T and P are as above.

NOTE.—In the case of front plates of boilers in the steam space, these numbers should be reduced 20 per cent., unless the plates are guarded from the direct action of the heat.

49. For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per sq in.};$$

where T = the thickness of the plates in sixteenths of an inch,

P = the *mean* pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per sq. in.};$$

where P = the horizontal distance from centre to centre of the bounding rows of tubes, and

C = 120 where the stay tubes are pitched with two plain tubes between them and are not fitted with nuts outside the plates,

C = 130 if they are fitted with nuts outside the plates.

C = 140 if each alternate tube is a stay tube not fitted with nuts,

C = 150 if they are fitted with nuts outside the plates,

C = 160 if every tube in these rows is a stay tube and not fitted with nuts,

C = 170 if every tube in these rows is a stay tube and each alternate stay tube is fitted with nuts outside the plates.

50. The thickness of tube plates of Combustion Chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule:—

$$T = \frac{P \times W \times D}{1750 \times (D - d)}$$

where P = working pressure in lbs. per square inch.

W = width of Combustion Chamber between plates in inches.

D = horizontal pitch of tubes in inches.

d = inside diameter of plain tubes in inches.

T = thickness of tube plates in sixteenths of an inch.

Girders.

51. The strength of girders supporting the tops of combustion chambers and other flat surfaces to be taken from the following formula:—

$$\frac{C \times d^2 \times T}{(L - P) \times D \times L} = \text{working pressure in lbs. per sq. in.};$$

where L = width between tube plates, or tube plate and back plate of chamber

P = pitch of stays in girders,

D = distance from centre to centre of girders,

d = depth of girder at centre,

T = thickness of girder at centre. All these dimensions to be taken in inches.

Wrought Iron.

C = {
- 6000, if there is one stay to each girder.
- 9000, if there are two or three stays to each girder.
- 10000, if there are four or five stays to each girder.
- 10500, if there are six or seven stays to each girder.
- 10800, if there are eight stays or above to each girder.

Wrought Steel.

C = {
- 6600, if there is one stay to each girder.
- 9900, if there are two or three stays to each girder.
- 11000, if there are four or five stays to each girder.
- 11550, if there are six or seven stays to each girder.
- 11880, if there are eight stays or above to each girder.

Circular Furnaces.

52. The strength of plain furnaces to resist collapsing to be calculated as follows:—

Where the length of the plain cylindrical part of the furnace exceeds 120 times the thickness of the plate, the working pressure is to be calculated by the following formula:—

$$\frac{1075200 \times T^2}{L \times D} = \text{working pressure in lbs. per sq. in.};$$

Where the length of the plain cylindrical part of the furnace is less than 120 times the thickness of the plate, the working pressure is to be calculated by the following formula:—

$$\frac{50 \times (300\,T - L)}{D} = \text{working pressure in lbs. per sq. in.};$$

where D = outside diameter of furnace in inches,

T = thickness of plates in inches,

L = length of plain cylindrical part in inches, measured from the centres of the rivets connecting the furnaces to the flanges of the end and tube plates, or from the commencement of the curvature of the flanges of the furnace where it is flanged or fitted with Adamson rings.

53. In the furnaces referred to below the formulæ given are applicable if the steel used has a tensile strength of not less than 26 nor more than 30 tons per square inch. If the material of furnaces has a less tensile strength than 26 tons per square inch, then for each ton per square inch which the minimum tensile strength falls below 26, the co-efficient is to be correspondingly decreased by $\frac{1}{26}$th part.

54. The strength of corrugated furnaces made on Fox's Morison's, Deighton's, or Beardmore's plan, to be calculated from

$$\frac{1259 \times (T - 2)}{D} = \text{working pressure in lbs. per sq. in.}$$

55. The strength of spirally corrugated furnaces is to be calculated from the following formula:—

$$\frac{912 \times (T - 2)}{D} = \text{working pressure in lbs. per sq. in.;}$$

where T = thickness of plate in sixteenths of an inch,
and D = outside diameter of corrugated furnaces in inches.

56. The strength of Improved Purves' furnaces with ribs 9 inches apart, and of Brown's Cambered furnaces with ribs either 8 inches or 9 inches apart, to be calculated from the following formula:—

$$\frac{1160 \times (T - 2)}{D} = \text{working pressure in lbs. per sq. in.}$$

where T = thickness of plate in sixteenths of an inch,
and D = smallest diameter of furnaces, in inches.

57. The strength of the Leeds Forge bulb furnace is to be calculated from the following formula:—

$$\frac{1259 \times (T - 2)}{D} = \text{working pressure in lbs. per sq. in.;}$$

where T = thickness of plate in sixteenths of an inch,
and D = smallest outside diameter in inches.

58. The strength of Holmes' patent furnaces, in which the corrugations are not more than 16 inches apart from centre to centre, and not less than 2 inches high, to be calculated from the following formula:—

$$\text{Working pressure in lbs. per sq. in.} = \frac{945 \times (T - 2)}{D}$$

where T = thickness of plain portions of furnace in sixteenths of an inch,
and D = outside diameter of plain parts of the furnace in inches.

Donkey Boilers.

59. The iron used in the construction of the fire boxes, uptakes, and water tubes of donkey boilers shall be of good quality, and to the satisfaction of the Surveyors, who may in any cases where they deem it advisable apply the following tests:—

Thickness of Plates.	To Bend cold through an angle of	
	With the Grain.	Across the Grain.
	Degrees.	Degrees.
$\frac{5}{16}$	80	45
$\frac{6}{16}$	70	35
$\frac{7}{16}$	55	25
$\frac{8}{16}$	40	20

The material to stand bending *hot* to an angle of 90 degrees, over a radius not greater than $1\frac{1}{2}$ times the thickness of the plates.

Rules for Determining Sizes of Shafts.

60. The diameters of intermediate shafts are to be not less than those given by the following formula:—

For Compound Engines with two cranks at right angles—

Diameter of intermediate shaft in inches =

$$(\cdot 04\ A + \cdot 006\ D + \cdot 02\ S) \times \sqrt[3]{P}$$

For Triple expansion engines with three cranks at equal angles—

Diameter of intermediate shaft in inches =

$$(\cdot 038\ A + \cdot 009\ B + \cdot 002\ D + \cdot 0165\ S) \times \sqrt[3]{P}$$

For Quadruple expansion engines with two cranks at right angles—

Diameter of intermediate shaft in inches =

$$(\cdot 034\ A + \cdot 011\ B + \cdot 004\ C + \cdot 0014 D + \cdot 016\ S) \times \sqrt[3]{P}$$

For Quadruple expansion engines with three cranks—

Diameter of intermediate shaft in inches =

$$(\cdot 028\ A + \cdot 014\ B + \cdot 006\ C + \cdot 0017\ D + \cdot 015\ S) \times \sqrt[3]{P}$$

For Quadruple expansion engines with four cranks—

Diameter of intermediate shaft in inches =

$$(\cdot 033\ A + \cdot 01\ B + \cdot 004\ C + \cdot 0013\ D + \cdot 0155\ S) \times \sqrt[3]{P}$$

where A = diameter of High Pressure Cylinders in inches,

B = diameter of first Intermediate Cylinder in inches,

C = diameter of second Intermediate Cylinder in inches,

D = diameter of Low Pressure Cylinder in inches,

S = Stroke of Pistons in inches,

P = Boiler pressure above atmosphere in lbs. per square inch.

61. The diameter of crank shaft, and of thrust shaft under the collars, to be at least $\frac{21}{20}$ths of that of the intermediate shaft. The diameter of thrust shaft may be tapered off at each end to the same size as that of the intermediate shaft.

62. The diameter of the screw shaft to be equal to the diameter of intermediate shaft (found as above) multiplied by $\left(\cdot 63 + \frac{\cdot 03\ P}{T}\right)$, but in no case to be less than $1 \cdot 07$ T.

where P is the diameter of propeller, and

T the diameter of intermediate shaft, both in inches.

The size of screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in paragraph 32. If no liners are used or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$ths that given above.

The diameter of screw shaft is to be tapered off at the forward end to the size of the crank shaft.

63. Note.—The Rules are intended to apply to Two Cylinder Compound Engines, in which the ratio of areas of Low and High Pressure Cylinders does not exceed 4·5 to 1; to Triple Expansion Engines in which it does not exceed 9 to 1; to Quadruple Expansion Engines in which it does not exceed 12 to 1; and in all cases, as regards the stroke, in which the length of stroke is not less than one-half the diameter or greater than the diameter of the Low Pressure Cylinder. Engines of extreme proportions beyond these limits being specially submitted to be dealt with on their merits.

Areas of Circles.

Advancing by Thirty-Seconds.

Diam	0	1	2	3	4	5	6	Diam
		·78540	3·1416	7·0686	12·566	19·635	28·274	
$\frac{1}{32}$	·000767	·83525	3·2405	7·2166	12·763	19·881	28·570	$\frac{1}{32}$
$\frac{1}{16}$	·003068	·88664	3·3410	7·3662	12·962	20·129	28·866	$\frac{1}{16}$
$\frac{3}{32}$	·006903	·93956	3·4430	7·5173	13·162	20·378	29·165	$\frac{3}{32}$
$\frac{1}{8}$	·012272	·99402	3·5466	7·6699	13·364	20·629	29·465	$\frac{1}{8}$
$\frac{5}{32}$	·019175	1·0500	3·6516	7·8241	13·567	20·881	29·766	$\frac{5}{32}$
$\frac{3}{16}$	·027612	1·1075	3·7583	7·9798	13·772	21·135	30·069	$\frac{3}{16}$
$\frac{7}{32}$	·037583	1·1666	3·8664	8·1370	13·978	21·391	30·374	$\frac{7}{32}$
$\frac{1}{4}$	·049087	1·2272	3·9761	8·2958	14·186	21·648	30·680	$\frac{1}{4}$
$\frac{9}{32}$	·062126	1·2893	4·0873	8·4561	14·396	21·906	30·987	$\frac{9}{32}$
$\frac{5}{16}$	·076699	1·3530	4·2000	8·6179	14·607	22·166	31·296	$\frac{5}{16}$
$\frac{11}{32}$	·092806	1·4182	4·3143	8·7813	14·819	22·428	31·607	$\frac{11}{32}$
$\frac{3}{8}$	·11045	1·4849	4·4301	8·9462	15·033	22·691	31·919	$\frac{3}{8}$
$\frac{13}{32}$	·12962	1·5532	4·5475	9·1126	15·249	22·955	32·233	$\frac{13}{32}$
$\frac{7}{16}$	·15033	1·6230	4·6664	9·2806	15·466	23·221	32·548	$\frac{7}{16}$
$\frac{15}{32}$	·17257	1·6943	4·7868	9·4501	15·684	23·489	32·865	$\frac{15}{32}$
$\frac{1}{2}$	·19635	1·7671	4·9087	9·6211	15·904	23·758	33·183	$\frac{1}{2}$
$\frac{17}{32}$	·22166	1·8415	5·0322	9·7937	16·126	24·029	33·503	$\frac{17}{32}$
$\frac{9}{16}$	·24850	1·9175	5·1572	9·9678	16·349	24·301	33·824	$\frac{9}{16}$
$\frac{19}{32}$	·27688	1·9949	5·2838	10·143	16·574	24·575	34·147	$\frac{19}{32}$
$\frac{5}{8}$	·30680	2·0739	5·4119	10·321	16·800	24·850	34·472	$\frac{5}{8}$
$\frac{21}{32}$	·33824	2·1545	5·5415	10·499	17·028	25·127	34·798	$\frac{21}{32}$
$\frac{11}{16}$	·37122	2·2365	5·6727	10·680	17·257	25·406	35·125	$\frac{11}{16}$
$\frac{23}{32}$	·40574	2·3201	5·8054	10·861	17·488	25·686	35·454	$\frac{23}{32}$
$\frac{3}{4}$	·44179	2·4053	5·9396	11·045	17·721	25·967	35·785	$\frac{3}{4}$
$\frac{25}{32}$	·47937	2·4920	6·0753	11·230	17·954	26·250	36·117	$\frac{25}{32}$
$\frac{13}{16}$	·51849	2·5802	6·2126	11·416	18·190	26·535	36·450	$\frac{13}{16}$
$\frac{27}{32}$	·55914	2·6699	6·3514	11·604	18·427	26·821	36·786	$\frac{27}{32}$
$\frac{7}{8}$	·60132	2·7612	6·4918	11·793	18·665	27·109	37·122	$\frac{7}{8}$
$\frac{29}{32}$	·64504	2·8540	6·6337	11·984	18·906	27·398	37·461	$\frac{29}{32}$
$\frac{15}{16}$	·69029	2·9483	6·7771	12·177	19·147	27·688	37·800	$\frac{15}{16}$
$\frac{31}{32}$	·73708	3·0442	6·9221	12·371	19·390	27·981	38·142	$\frac{31}{32}$

Areas of Circles.

Advancing by Thirty-Seconds.

Diam	7	8	9	10	11	12	13	Diam
	38·485	50·265	63·617	78·540	95·033	113·10	132·73	
1/32	38·829	50·659	64·060	79·031	95·574	113·69	133·37	1/32
1/16	39·175	51·054	64·504	79·525	96·116	114·28	134·01	1/16
3/32	39·522	51·450	64·950	80·019	96·660	114·87	134·65	3/32
1/8	39·871	51·849	65·397	80·516	97·205	115·47	135·30	1/8
5/32	40·222	52·248	65·845	81·013	97·752	116·06	135·94	5/32
3/16	40·574	52·649	66·296	81·513	98·301	116·66	136·59	3/16
7/32	40·927	53·052	66·747	82·014	98·850	117·26	137·24	7/32
1/4	41·282	53·456	67·201	82·516	99·402	117·86	137·89	1/4
9/32	41·639	53·862	67·655	83·020	99·955	118·46	138·54	9/32
5/16	41·997	54·269	68·112	83·525	100·51	119·06	139·19	5/16
11/32	42·357	54·678	68·570	84·032	101·07	119·67	139·84	11/32
3/8	42·718	55·088	69·029	84·541	101·62	120·28	140·50	3/8
13/32	43·081	55·500	69·490	85·051	102·18	120·88	141·16	13/32
7/16	43·445	55·914	69·953	85·562	102·74	121·49	141·82	7/16
15/32	43·811	56·329	70·417	86·076	103·31	122·11	142·48	15/32
1/2	44·179	56·745	70·882	86·590	103·87	122·72	143·14	1/2
17/32	44·548	57·163	71·349	87·106	104·43	123·33	143·80	17/32
9/16	44·918	57·583	71·818	87·624	105·00	123·95	144·47	9/16
19/32	45·290	58·004	72·288	88·143	105·57	124·57	145·13	19/32
5/8	45·664	58·426	72·760	88·664	106·14	125·19	145·80	5/8
21/32	46·039	58·850	73·233	89·186	106·71	125·81	146·47	21/32
11/16	46·415	59·276	73·708	89·710	107·28	126·43	147·14	11/16
23/32	46·793	59·703	74·184	90·236	107·86	127·05	147·82	23/32
3/4	47·173	60·132	74·662	90·763	108·43	127·68	148·49	3/4
25/32	47·554	60·562	75·141	91·291	109·01	128·30	149·17	25/32
13/16	47·937	60·994	75·622	91·821	109·59	128·93	149·84	13/16
27/32	48·321	61·427	76·105	92·353	110·17	129·56	150·52	27/32
7/8	48·707	61·862	76·589	92·886	110·75	130·19	151·20	7/8
29/32	49·094	62·299	77·074	93·420	111·34	130·82	151·88	29/32
15/16	49·483	62·737	77·561	93·956	111·92	131·46	152·57	15/16
31/32	49·874	63·176	78·050	94·494	112·51	132·09	153·25	31/32

Areas of Circles.

Advancing by Thirty-Seconds.

Diam	14	15	16	17	18	19	20	Diam
	153·94	176·71	201·06	226·98	254·47	283·53	314·16	
$\frac{1}{32}$	154·63	177·45	201·85	227·82	255·35	284·46	315·14	$\frac{1}{32}$
$\frac{1}{16}$	155·32	178·19	202·64	228·65	256·24	285·40	316·13	$\frac{1}{16}$
$\frac{3}{32}$	156·01	178·93	203·43	229·49	257·13	286·33	317·11	$\frac{3}{32}$
$\frac{1}{8}$	156·70	179·67	204·22	230·33	258·02	287·27	318·10	$\frac{1}{8}$
$\frac{5}{32}$	157·39	180·42	205·01	231·17	258·91	288·21	319·09	$\frac{5}{32}$
$\frac{3}{16}$	158·09	181·16	205·80	232·01	259·80	289·15	320·08	$\frac{3}{16}$
$\frac{7}{32}$	158·79	181·91	206·60	232·86	260·69	290·09	321·07	$\frac{7}{32}$
$\frac{1}{4}$	159·48	182·65	207·39	233·71	261·59	291·04	322·06	$\frac{1}{4}$
$\frac{9}{32}$	160·19	183·40	208·19	234·55	262·48	291·98	323·06	$\frac{9}{32}$
$\frac{5}{16}$	160·89	184·15	208·99	235·40	263·38	292·93	324·05	$\frac{5}{16}$
$\frac{11}{32}$	161·59	184·91	209·79	236·25	264·28	293·88	325·05	$\frac{11}{32}$
$\frac{3}{8}$	162·30	185·66	210·60	237·10	265·18	294·83	326·05	$\frac{3}{8}$
$\frac{13}{32}$	163·00	186·42	211·40	237·96	266·09	295·78	327·05	$\frac{13}{32}$
$\frac{7}{16}$	163·71	187·17	212·21	238·81	266·99	296·74	328·05	$\frac{7}{16}$
$\frac{15}{32}$	164·42	187·93	213·02	239·67	267·90	297·69	329·06	$\frac{15}{32}$
$\frac{1}{2}$	165·13	188·69	213·82	240·53	268·80	298·65	330·06	$\frac{1}{2}$
$\frac{17}{32}$	165·84	189·45	214·64	241·39	269·71	299·61	331·07	$\frac{17}{32}$
$\frac{9}{16}$	166·56	190·22	215·45	242·25	270·62	300·57	332·08	$\frac{9}{16}$
$\frac{19}{32}$	167·27	190·98	216·26	243·11	271·53	301·53	333·09	$\frac{19}{32}$
$\frac{5}{8}$	167·99	191·75	217·08	243·98	272·45	302·49	334·10	$\frac{5}{8}$
$\frac{21}{32}$	168·71	192·52	217·89	244·84	273·36	303·45	335·11	$\frac{21}{32}$
$\frac{11}{16}$	169·43	193·28	218·71	245·71	274·28	304·42	336·13	$\frac{11}{16}$
$\frac{23}{32}$	170·15	194·06	219·53	246·58	275·20	305·39	337·15	$\frac{23}{32}$
$\frac{3}{4}$	170·87	194·83	220·35	247·45	276·12	306·35	338·16	$\frac{3}{4}$
$\frac{25}{32}$	171·60	195·60	221·18	248·32	277·04	307·32	339·18	$\frac{25}{32}$
$\frac{13}{16}$	172·32	196·38	222·00	249·20	277·96	308·30	340·20	$\frac{13}{16}$
$\frac{27}{32}$	173·05	197·15	222·83	250·07	278·88	309·27	341·23	$\frac{27}{32}$
$\frac{7}{8}$	173·78	197·93	223·65	250·95	279·81	310·24	342·25	$\frac{7}{8}$
$\frac{29}{32}$	174·51	198·71	224·48	251·83	280·74	311·22	343·28	$\frac{29}{32}$
$\frac{15}{16}$	175·25	199·49	225·31	252·70	281·67	312·20	344·30	$\frac{15}{16}$
$\frac{31}{32}$	175·98	200·28	226·15	253·59	282·60	313·18	345·33	$\frac{31}{32}$

Areas of Circles.

Advancing by Sixteenths.

Diam	21	22	23	24	25	26	27	Diam
	346·36	380·13	415·48	452·39	490·87	530·93	572·56	
$\frac{1}{16}$	348·43	382·30	417·74	454·75	493·33	533·48	575·21	$\frac{1}{16}$
$\frac{1}{8}$	350·50	384·46	420·00	457·11	495·79	536·05	577·87	$\frac{1}{8}$
$\frac{3}{16}$	352·57	386·64	422·28	459·49	498·26	538·61	580·54	$\frac{3}{16}$
$\frac{1}{4}$	354·66	388·82	424·56	461·86	500·74	541·19	583·21	$\frac{1}{4}$
$\frac{5}{16}$	356·75	391·01	426·84	464·25	503·22	543·77	585·89	$\frac{5}{16}$
$\frac{3}{8}$	358·84	393·20	429·13	466·64	505·71	546·35	588·57	$\frac{3}{8}$
$\frac{7}{16}$	360·94	395·40	431·43	469·03	508·20	548·95	591·26	$\frac{7}{16}$
$\frac{1}{2}$	363·05	397·61	433·74	471·44	510·71	551·55	593·96	$\frac{1}{2}$
$\frac{9}{16}$	365·16	399·82	436·05	473·84	513·21	554·15	596·66	$\frac{9}{16}$
$\frac{5}{8}$	367·28	402·04	438·36	476·26	515·72	556·76	599·37	$\frac{5}{8}$
$\frac{11}{16}$	369·41	404·26	440·69	478·68	518·24	559·38	602·08	$\frac{11}{16}$
$\frac{3}{4}$	371·54	406·49	443·01	481·11	520·77	562·00	604·81	$\frac{3}{4}$
$\frac{13}{16}$	373·68	408·73	445·35	483·54	523·30	564·63	607·53	$\frac{13}{16}$
$\frac{7}{8}$	375·83	410·97	447·69	485·98	525·84	567·27	610·27	$\frac{7}{8}$
$\frac{15}{16}$	377·98	413·22	450·04	488·42	528·38	569·91	613·01	$\frac{15}{16}$

Diam	28	29	30	31	32	33	34	Diam
	615·75	660·52	706·86	754·77	804·25	855·30	907·92	
$\frac{1}{16}$	618·50	663·37	709·81	757·81	807·39	858·54	911·26	$\frac{1}{16}$
$\frac{1}{8}$	621·26	666·23	712·76	760·87	810·54	861·79	914·61	$\frac{1}{8}$
$\frac{3}{16}$	624·03	669·09	715·72	763·93	813·70	865·05	917·96	$\frac{3}{16}$
$\frac{1}{4}$	626·80	671·96	718·69	766·99	816·86	868·31	921·32	$\frac{1}{4}$
$\frac{5}{16}$	629·57	674·83	721·66	770·06	820·03	871·57	924·69	$\frac{5}{16}$
$\frac{3}{8}$	632·36	677·71	724·64	773·14	823·21	874·85	928·06	$\frac{3}{8}$
$\frac{7}{16}$	635·14	680·60	727·63	776·22	826·39	878·13	931·44	$\frac{7}{16}$
$\frac{1}{2}$	637·94	683·49	730·62	779·31	829·58	881·41	934·82	$\frac{1}{2}$
$\frac{9}{16}$	640·74	686·39	733·61	782·41	832·77	884·71	938·21	$\frac{9}{16}$
$\frac{5}{8}$	643·55	689·30	736·62	785·51	835·97	888·00	941·61	$\frac{5}{8}$
$\frac{11}{16}$	646·36	692·21	739·63	788·62	839·18	891·31	945·01	$\frac{11}{16}$
$\frac{3}{4}$	649·18	695·13	742·64	791·73	842·39	894·62	948·42	$\frac{3}{4}$
$\frac{13}{16}$	652·01	698·05	745·67	794·85	845·61	897·93	951·83	$\frac{13}{16}$
$\frac{7}{8}$	654·84	700·98	748·69	797·98	848·83	901·26	955·25	$\frac{7}{8}$
$\frac{15}{16}$	657·68	703·92	751·73	801·11	852·06	904·59	958·68	$\frac{15}{16}$

Areas of Circles.

Advancing by Sixteenths.

Diam	35	36	37	38	39	40	41	Diam
	962·11	1017·9	1075·2	1134·1	1194·6	1256·6	1320·3	
$\frac{1}{16}$	965·55	1021·4	1078·8	1137·8	1198·4	1260·6	1324·3	$\frac{1}{16}$
$\frac{1}{8}$	969·00	1025·0	1082·5	1141·6	1202·3	1264·5	1328·3	$\frac{1}{8}$
$\frac{3}{16}$	972·45	1028·5	1086·1	1145·3	1206·1	1268·4	1332·4	$\frac{3}{16}$
$\frac{1}{4}$	975·91	1032·1	1089·8	1149·1	1210·0	1272·4	1336·4	$\frac{1}{4}$
$\frac{5}{16}$	979·37	1035·6	1093·4	1152·8	1213·8	1276·3	1340·5	$\frac{5}{16}$
$\frac{3}{8}$	982·84	1039·2	1097·1	1156·6	1217·7	1280·3	1344·5	$\frac{3}{8}$
$\frac{7}{16}$	986·32	1042·8	1100·8	1160·4	1221·5	1284·3	1348·6	$\frac{7}{16}$
$\frac{1}{2}$	989·80	1046·3	1104·5	1164·2	1225·4	1288·2	1352·7	$\frac{1}{2}$
$\frac{9}{16}$	993·29	1049·9	1108·2	1167·9	1229·3	1292·2	1356·7	$\frac{9}{16}$
$\frac{5}{8}$	996·78	1053·5	1111·8	1171·7	1233·2	1296·2	1360·8	$\frac{5}{8}$
$\frac{11}{16}$	1000·3	1057·1	1115·5	1175·5	1237·1	1300·2	1364·9	$\frac{11}{16}$
$\frac{3}{4}$	1003·8	1060·7	1119·2	1179·3	1241·0	1304·2	1369·0	$\frac{3}{4}$
$\frac{13}{16}$	1007·3	1064·3	1123·0	1183·1	1244·9	1308·2	1373·1	$\frac{13}{16}$
$\frac{7}{8}$	1010·8	1068·0	1126·7	1186·9	1248·8	1312·2	1377·2	$\frac{7}{8}$
$\frac{15}{16}$	1014·3	1071·6	1130·4	1190·8	1252·7	1316·2	1381·3	$\frac{15}{16}$
Diam	**42**	**43**	**44**	**45**	**46**	**47**	**48**	**Diam**
	1385·4	1452·2	1520·5	1590·4	1661·9	1734·9	1809·6	
$\frac{1}{16}$	1389·6	1456·4	1524·9	1594·9	1666·4	1739·6	1814·3	$\frac{1}{16}$
$\frac{1}{8}$	1393·7	1460·7	1529·2	1599·3	1670·9	1744·2	1819·0	$\frac{1}{8}$
$\frac{3}{16}$	1397·8	1464·9	1533·5	1603·7	1675·5	1748·8	1823·7	$\frac{3}{16}$
$\frac{1}{4}$	1402·0	1469·1	1537·9	1608·2	1680·0	1753·5	1828·5	$\frac{1}{4}$
$\frac{5}{16}$	1406·1	1473·4	1542·2	1612·6	1684·6	1758·1	1833·2	$\frac{5}{16}$
$\frac{3}{8}$	1410·3	1477·6	1546·6	1617·0	1689·1	1762·7	1837·9	$\frac{3}{8}$
$\frac{7}{16}$	1414·5	1481·9	1550·9	1621·5	1693·7	1767·4	1842·7	$\frac{7}{16}$
$\frac{1}{2}$	1418·6	1486·2	1555·3	1626·0	1698·2	1772·1	1847·5	$\frac{1}{2}$
$\frac{9}{16}$	1422·8	1490·4	1559·7	1630·4	1702·8	1776·7	1852·2	$\frac{9}{16}$
$\frac{5}{8}$	1427·0	1494·7	1564·0	1634·9	1707·4	1781·4	1857·0	$\frac{5}{8}$
$\frac{11}{16}$	1431·2	1499·0	1568·4	1639·4	1712·0	1786·1	1861·8	$\frac{11}{16}$
$\frac{3}{4}$	1435·4	1503·3	1572·8	1643·9	1716·5	1790·8	1866·5	$\frac{3}{4}$
$\frac{13}{16}$	1439·6	1507·6	1577·2	1648·4	1721·1	1795·4	1871·3	$\frac{13}{16}$
$\frac{7}{8}$	1443·8	1511·9	1581·6	1652·9	1725·7	1800·1	1876·1	$\frac{7}{8}$
$\frac{15}{16}$	1448·0	1516·2	1586·0	1657·4	1730·3	1804·8	1880·9	$\frac{15}{16}$

Areas of Circles.

Advancing by Eighths.

Diam	49	50	51	52	53	54	55	Diam
	1885·7	1963·5	2042·8	2123·7	2206·2	2290·2	2375·8	
1/8	1895·4	1973·3	2052·8	2133·9	2216·6	2300·8	2386·6	1/8
1/4	1905·0	1983·2	2062·9	2144·2	2227·0	2311·5	2397·5	1/4
3/8	1914·7	1993·1	2073·0	2154·5	2237·5	2322·1	2408·3	3/8
1/2	1924·4	2003·0	2083·1	2164·8	2248·0	2332·8	2419·2	1/2
5/8	1934·2	2012·9	2093·2	2175·1	2258·5	2343·5	2430·1	5/8
3/4	1943·9	2022·8	2103·3	2185·4	2269·1	2354·3	2441·1	3/4
7/8	1953·7	2032·8	2113·5	2195·8	2279·6	2365·0	2452·0	7/8
Diam	56	57	58	59	60	61	62	Diam
	2463·0	2551·8	2642·1	2734·0	2827·4	2922·5	3019·1	
1/8	2474·0	2563·0	2653·5	2745·6	2839·2	2934·5	3031·3	1/8
1/4	2485·0	2574·2	2664·9	2757·2	2851·0	2946·5	3043·5	1/4
3/8	2496·1	2585·4	2676·4	2768·8	2862·9	2958·5	3055·7	3/8
1/2	2507·2	2596·7	2687·8	2780·5	2874·8	2970·6	3068·0	1/2
5/8	2518·3	2608·0	2699·3	2792·2	2886·6	2982·7	3080·2	5/8
3/4	2529·4	2619·4	2710·9	2803·9	2898·6	2994·8	3092·6	3/4
7/8	2540·6	2630·7	2722·4	2815·7	2910·5	3006·9	3104·9	7/8
Diam	63	64	65	66	67	68	69	Diam
	3117·2	3217·0	3318·3	3421·2	3525·7	3631·7	3739·3	
1/8	3129·6	3229·6	3331·1	3434·2	3538·8	3645·0	3752·8	1/8
1/4	3142·0	3242·2	3343·9	3447·2	3552·0	3658·4	3766·4	1/4
3/8	3154·5	3254·8	3356·7	3460·2	3565·2	3671·8	3780·0	3/8
1/2	3166·9	3267·5	3369·6	3473·2	3578·5	3685·3	3793·7	1/2
5/8	3179·4	3280·1	3382·4	3486·3	3591·7	3698·7	3807·3	5/8
3/4	3191·9	3292·8	3395·3	3499·4	3605·0	3712·2	3821·0	3/4
7/8	3204·4	3305·6	3408·2	3512·5	3618·3	3725·7	3834·7	7/8
Diam	70	71	72	73	74	75	76	Diam
	3848·5	3959·2	4071·5	4185·4	4300·8	4417·9	4536·5	
1/8	3862·2	3973·1	4085·7	4199·7	4315·4	4432·6	4551·4	1/8
1/4	3876·0	3987·1	4099·8	4214·1	4329·9	4447·4	4566·4	1/4
3/8	3889·8	4001·1	4114·0	4228·5	4344·5	4462·2	4581·3	3/8
1/2	3903·6	4015·2	4128·2	4242·9	4359·2	4477·0	4596·3	1/2
5/8	3917·5	4029·2	4142·5	4257·4	4373·8	4491·8	4611·4	5/8
3/4	3931·4	4043·3	4156·8	4271·8	4388·5	4506·7	4626·4	3/4
7/8	3945·3	4057·4	4171·1	4286·3	4403·2	4521·5	4641·5	7/8

Areas of Circles.

Advancing by Eighths.

Diam	77	78	79	80	81	82	83	Diam
	4656·6	4778·4	4901·7	5026·5	5153·0	5281·0	5410·6	
1/8	4671·8	4793·7	4917·2	5042·3	5168·9	5297·1	5426·9	1/8
1/4	4686·9	4809·0	4932·7	5058·0	5184·9	5313·3	5443·3	1/4
3/8	4702·1	4824·4	4948·3	5073·8	5200·8	5329·4	5459·6	3/8
1/2	4717·3	4839·8	4963·9	5089·6	5216·8	5345·6	5476·0	1/2
5/8	4732·5	4855·2	4979·5	5105·4	5232·8	5361·8	5492·4	5/8
3/4	4747·8	4870·7	4995·2	5121·2	5248·9	5378·1	5508·8	3/4
7/8	4763·1	4886·2	5010·9	5137·1	5264·9	5394·3	5525·3	7/8

Diam	84	85	86	87	88	89	90	Diam
	5541·8	5674·5	5808·8	5944·7	6082·1	6221·1	6361·7	
1/8	5558·3	5691·2	5825·7	5961·8	6099·4	6238·6	6379·4	1/8
1/4	5574·8	5707·9	5842·6	5978·9	6116·7	6256·1	6397·1	1/4
3/8	5591·4	5724·7	5859·6	5996·0	6134·1	6273·7	6414·9	3/8
1/2	5607·9	5741·5	5876·5	6013·2	6151·4	6291·2	6432·6	1/2
5/8	5624·5	5758·3	5893·5	6030·4	6168·8	6308·8	6450·4	5/8
3/4	5641·2	5775·1	5910·6	6047·6	6186·2	6326·4	6468·2	3/4
7/8	5657·8	5791·9	5927·6	6064·9	6203·7	6344·1	6486·0	7/8

Diam	91	92	93	94	95	96	97	Diam
	6503·9	6647·6	6792·9	6939·8	7088·2	7238·2	7389·8	
1/8	6521·8	6665·7	6811·2	6958·2	7106·9	7257·1	7408·9	1/8
1/4	6539·7	6683·8	6829·5	6976·7	7125·6	7276·0	7428·0	1/4
3/8	6557·6	6701·9	6847·8	6995·3	7144·3	7294·9	7447·1	3/8
1/2	6575·6	6720·1	6866·1	7013·8	7163·0	7313·8	7466·2	1/2
5/8	6593·5	6738·2	6884·5	7032.4	7181·8	7332·8	7485·3	5/8
3/4	6611·5	6756·4	6902·9	7051 0	7200·6	7351·8	7504·5	3/4
7/8	6629·6	6774·7	6921·3	7069·6	7219·4	7370·8	7523·7	7/8

Diam	98	99	100	101	102	103	104	Diam
	7543·0	7697·7	7854·0	8011·8	8171·3	8332·3	8494·9	
1/8	7562·2	7717·1	7873·6	8031·7	8191·3	8352·5	8515·3	1/8
1/4	7581·5	7736·6	7893·3	8051·6	8211·4	8372·8	8535·8	1/4
3/8	7600·8	7756·1	7913·0	8071·5	8231·5	8393·1	8556·2	3/8
1/2	7620·1	7775·6	7932·7	8091·4	8251·6	8413·4	8576·7	1/2
5/8	7639·5	7795·2	7952·5	8111·3	8271·7	8433·7	8597·3	5/8
3/4	7658·9	7814·8	7972·2	8131·3	8291·9	8454·1	8617·8	3/4
7/8	7678·3	7834·4	7992·0	8151·3	8312·1	8474·5	8638·4	7/8

Circumferences of Circles.

Advancing by Thirty-Seconds.

Fractions.	0	1	2	3	4	5	6	7
	...	3·1416	6·2832	9·4248	12·566	15·708	18·850	21·991
$\frac{1}{32}$	·09817	3·2398	6·3814	9·5230	12·665	15·806	18·948	22·089
$\frac{1}{16}$	·19635	3·3379	6·4795	9·6211	12·763	15·904	19·046	22·187
$\frac{3}{32}$	·29452	3·4361	6·5777	9·7193	12·861	16·002	19·144	22·286
$\frac{1}{8}$	·39270	3·5343	6·6759	9·8175	12·959	16·101	19·242	22·384
$\frac{5}{32}$	·49087	3·6325	6·7741	9·9157	13·057	16·199	19·340	22·482
$\frac{3}{16}$	·58905	3·7306	6·8722	10·014	13·155	16·297	19·439	22·580
$\frac{7}{32}$	·68722	3·8288	6·9704	10·112	13·254	16·395	19·537	22·678
$\frac{1}{4}$	·78540	3·9270	7·0686	10·210	13·352	16·493	19·635	22·777
$\frac{9}{32}$	·88357	4·0252	7·1668	10·308	13·450	16·592	19·733	22·875
$\frac{5}{16}$	·98175	4·1233	7·2649	10·407	13·548	16·690	19·831	22·973
$\frac{11}{32}$	1·0799	4·2215	7·3631	10·505	13·646	16·788	19·929	23·071
$\frac{3}{8}$	1·1781	4·3197	7·4613	10·603	13·744	16·886	20·028	23·169
$\frac{13}{32}$	1·2763	4·4179	7·5595	10·701	13·843	16·984	20·126	23·267
$\frac{7}{16}$	1·3744	4·5160	7·6576	10·799	13·941	17·082	20·224	23·366
$\frac{15}{32}$	1·4726	4·6142	7·7558	10·897	14·039	17·181	20·322	23·464
$\frac{1}{2}$	1·5708	4·7124	7·8540	10·996	14·137	17·279	20·420	23·562
$\frac{17}{32}$	1·6690	4·8106	7·9522	11·094	14·235	17·377	20·519	23·660
$\frac{9}{16}$	1·7671	4·9087	8·0503	11·192	14·334	17·475	20·617	23·758
$\frac{19}{32}$	1·8653	5·0069	8·1485	11·290	14·432	17·573	20·715	23·856
$\frac{5}{8}$	1·9635	5·1051	8·2467	11·388	14·530	17·671	20·813	23·955
$\frac{21}{32}$	2·0617	5·2033	8·3449	11·486	14·628	17·770	20·911	24·053
$\frac{11}{16}$	2·1598	5·3014	8·4430	11·585	14·726	17·868	21·009	24·151
$\frac{23}{32}$	2·2580	5·3996	8·5412	11·683	14·824	17·966	21·108	24·249
$\frac{3}{4}$	2·3562	5·4978	8·6394	11·781	14·923	18·064	21·206	24·347
$\frac{25}{32}$	2·4544	5·5960	8·7376	11·879	15·021	18·162	21·304	24·446
$\frac{13}{16}$	2·5525	5·6941	8·8357	11·977	15·119	18·261	21·402	24·544
$\frac{27}{32}$	2·6507	5·7923	8·9339	12·075	15·217	18·359	21·500	24·642
$\frac{7}{8}$	2·7489	5·8905	9·0321	12·174	15·315	18·457	21·598	24·740
$\frac{29}{32}$	2·8471	5·9887	9·1303	12·272	15·413	18·555	21·697	24·838
$\frac{15}{16}$	2·9452	6·0868	9·2284	12·370	15·512	18·653	21·795	24·936
$\frac{31}{32}$	3·0434	6·1850	9·3266	12·468	15·610	18·751	21·893	25·035

INDEX.

Areas of Circles.

Advancing by Thirty-Seconds.

Diam	14	15	16	17	18	19	20	Diam
	153·94	176·71	201·06	226·98	254·47	283·53	314·16	
$\frac{1}{32}$	154·63	177·45	201·85	227·82	255·35	284·46	315·14	$\frac{1}{32}$
$\frac{1}{16}$	155·32	178·19	202·64	228·65	256·24	285·40	316·13	$\frac{1}{16}$
$\frac{3}{32}$	156·01	178·93	203·43	229·49	257·13	286·33	317·11	$\frac{3}{32}$
$\frac{1}{8}$	156·70	179·67	204·22	230·33	258·02	287·27	318·10	$\frac{1}{8}$
$\frac{5}{32}$	157·39	180·42	205·01	231·17	258·91	288·21	319·09	$\frac{5}{32}$
$\frac{3}{16}$	158·09	181·16	205·80	232·01	259·80	289·15	320·08	$\frac{3}{16}$
$\frac{7}{32}$	158·79	181·91	206·60	232·86	260·69	290·09	321·07	$\frac{7}{32}$
$\frac{1}{4}$	159·48	182·65	207·39	233·71	261·59	291·04	322·06	$\frac{1}{4}$
$\frac{9}{32}$	160·19	183·40	208·19	234·55	262·48	291·98	323·06	$\frac{9}{32}$
$\frac{5}{16}$	160·89	184·15	208·99	235·40	263·38	292·93	324·05	$\frac{5}{16}$
$\frac{11}{32}$	161·59	184·91	209·79	236·25	264·28	293·88	325·05	$\frac{11}{32}$
$\frac{3}{8}$	162·30	185·66	210·60	237·10	265·18	294·83	326·05	$\frac{3}{8}$
$\frac{13}{32}$	163·00	186·42	211·40	237·96	266·09	295·78	327·05	$\frac{13}{32}$
$\frac{7}{16}$	163·71	187·17	212·21	238·81	266·99	296·74	328·05	$\frac{7}{16}$
$\frac{15}{32}$	164·42	187·93	213·02	239·67	267·90	297·69	329·06	$\frac{15}{32}$
$\frac{1}{2}$	165·13	188·69	213·82	240·53	268·80	298·65	330·06	$\frac{1}{2}$
$\frac{17}{32}$	165·84	189·45	214·64	241·39	269·71	299·61	331·07	$\frac{17}{32}$
$\frac{9}{16}$	166·56	190·22	215·45	242·25	270·62	300·57	332·08	$\frac{9}{16}$
$\frac{19}{32}$	167·27	190·98	216·26	243·11	271·53	301·53	333·09	$\frac{19}{32}$
$\frac{5}{8}$	167·99	191·75	217·08	243·98	272·45	302·49	334·10	$\frac{5}{8}$
$\frac{21}{32}$	168·71	192·52	217·89	244·84	273·36	303·45	335·11	$\frac{21}{32}$
$\frac{11}{16}$	169·43	193·28	218·71	245·71	274·28	304·42	336·13	$\frac{11}{16}$
$\frac{23}{32}$	170·15	194·06	219·53	246·58	275·20	305·39	337·15	$\frac{23}{32}$
$\frac{3}{4}$	170·87	194·83	220·35	247·45	276·12	306·35	338·16	$\frac{3}{4}$
$\frac{25}{32}$	171·60	195·60	221·18	248·32	277·04	307·32	339·18	$\frac{25}{32}$
$\frac{13}{16}$	172·32	196·38	222·00	249·20	277·96	308·30	340·20	$\frac{13}{16}$
$\frac{27}{32}$	173·05	197·15	222·83	250·07	278·88	309·27	341·23	$\frac{27}{32}$
$\frac{7}{8}$	173·78	197·93	223·65	250·95	279·81	310·24	342·25	$\frac{7}{8}$
$\frac{29}{32}$	174·51	198·71	224·48	251·83	280·74	311·22	343·28	$\frac{29}{32}$
$\frac{15}{16}$	175·25	199·49	225·31	252·70	281·67	312·20	344·30	$\frac{15}{16}$
$\frac{31}{32}$	175·98	200·28	226·15	253·59	282·60	313·18	345·33	$\frac{31}{32}$

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